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Henry C. Tulley

Handbook on Engineering.

THE PRACTICAL CARE AND MANAGEMENT

OF

DYNAMOS, MOTORS, BOILERS, ENGINES, PUMPS, INSPIRATORS AND INJECTORS, REFRIGERATING MACHINERY, HYDRAULIC ELEVATORS, ELECTRIC ELEVATORS, AIR COMPRESSORS, ROPE TRANSMISSION AND ALL BRANCHES OF STEAM ENGINEERING.

BY

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INTRODUCTION.

The object of the Author in preparing this work has been to present to the practical engineer a book to which he can, with confidence, refer to for information regarding every branch of his profession.

Up to the date of the publication of this book, it was impossible to find a plain and practical treatise on the steam boiler, steam pump, steam engine, and dynamo, and how to care for them; electric and hydraulic elevators, and how to care for them; and all other work that an engineer is apt to come in contact with in his profession.

An experience of over twenty-five years with all kinds of engines and boilers, pumps, and all other kinds of machinery, enables the Author to fully understand the kind of information most needed by men having charge of steam engines of every description, and what they should comprehend and employ.

With this object in view, the Author has carefully made note of his past experience, and has also made note of things that came to his notice while visiting different engine rooms, and accordingly, has taken up each subject singly, excluding therefrom, everything not strictly connected with steam engineering.

Particular attention has been given to the latest improvements in all classes of steam engines, with rules and formulas according to the best modern practice, which, it is hoped, will be of great value to engineers, as nothing of the kind has heretofore been published.

This book also contains ample instructions for setting up, lining, reversing and setting the valves of all classes of engines.

THE AUTHOR.

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CHAPTER XXX.

HANDBOOK ON ENGINEERING.

CHAPTER I.

THE ELEMENTARY PRINCIPLES OF ELECTRICAL MACHINERY.

The operation of electric generators, or dynamos, as they are ordinarily called, and also that of electric motors, depends upon a simple relation between electricity and magnetism, which will be explained in a simple manner in the following paragraphs.

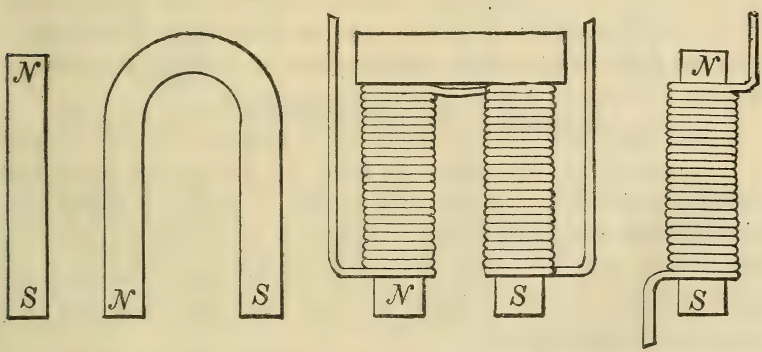


Fig. 1. Fig. 2. Fig. 3. Fig. 4.

Forms of magnets.

A permanent magnet, as is well known, is a bar of steel which possesses the power of attracting pieces of iron. These bars may be made straight, as in Fig. 1, or in the form of a U, as in Fig. 2, or in any other shape desired. The strength of a permanent magnet depends upon the kind of steel of which it is made, and

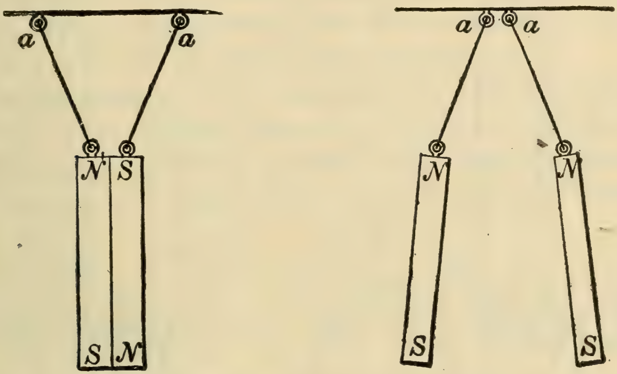
also upon the temper it is given. Generally speaking, the harder the steel the stronger the magnet. A bar of soft steel, or wrought iron, cannot be made into a permanent magnet of any noticeable strength, but if such a bar is covered with a coil of wire, as shown in Figs. 3 and 4, and a current of electricity is passed through the wire, the bar will be converted into a very strong magnet so long as the current flows. As soon as the electric current stops flowing through the wire, the magnetism of the bar will die out.

Magnets of the last-named type are called electro-magnets, as they do not possess magnet properties except when the electric current flows around them. Electro-magnets, when energized by sufficiently strong electric currents, can be far more powerful than the permanent magnets, and on that account they are used in electric generators and motors. In addition to being a stronger magnet, the electro-magnet has the advantage that it can be magnetized and demagnetized almost instantly, by simply cutting off the exciting electric current, and on this account they can be used for parts of electrical machines and apparatus, for which the permanent magnet would be entirely unsuited.

If we test the attractive power of a magnet, we will find that it is greatest at the ends, the force at the middle point being scarcely noticeable. A bar such as Fig. 1 or Fig. 3 might hold a piece of iron weighing several pounds, if presented to either end, while at the middle point, it might not be able to sustain more than an ounce or two. Owing to this fact, the ends are called the poles of the magnet.

When a magnet is suspended from its center, like a scale beam, and allowed to swing freely, it will be found that it will come to rest in a north and south position, and no matter how violently it may be moved around, it will always come to a state of rest with the same end pointing towards the north. On this account, the ends are called north and south poles, the north pole being the end that points toward the north.

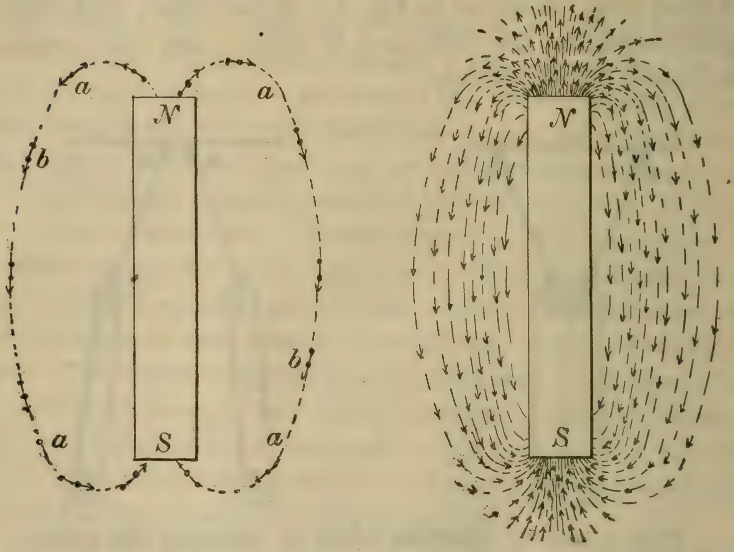
If two bar magnets are suspended side by side with the north end of one at the top and the north end of the other at the bottom, as illustrated in Fig. 5, they will attract each other; but if both magnets have the north end at the top, they will push away, as shown in Fig. 6. It is evident that there is a good reason for this difference in action, and this reason we can obtain by experiment.



Figs. 5 and 6. Showing effect of changing the poles.

A magnet needle, such as is used in the mariner's compass, is simply a small magnet. If we place a magnet bar, as shown in Fig. 7, and then set near to it, in different positions, a compass containing a very small needle, we will find that in these several positions the direction of the needle will be about as is indicated by the small arrows marked *b* on the curved lines *a a;* the arrow pointing towards the north end, or pole of the needle. The reason why the needle will take up these positions is that the north end of the bar attracts the south end of the needle, and pushes away the north end, just as in Figs. 5 and 6, and the south end of the bar acts in the same way; so that there is a tug of war going on, so to speak, between the attractions and repulsions of

the two ends of the bar upon the two ends of the needle, the result being that the position assumed by the needle is the resultant of these several forces. When the needle is near the



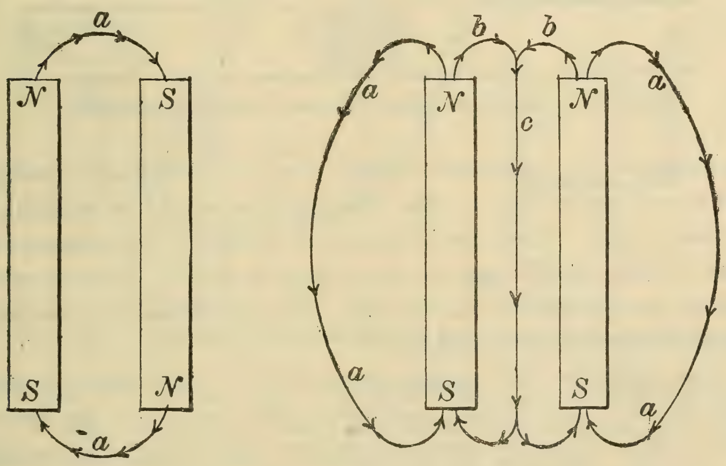
Figs. 7 and 8. Illustrating lines of force.

north pole of the bar, its south end is attracted with the greatest force, and when near the south end of the bar, the north end experiences the greatest attraction.

If we were to place the exploring needle in all possible positions near the magnet and trace lines parallel with it, in these positions, we would obtain a large number of curves about the shape of those shown in Fig. 8. As these curves represent the direction into which the magnet needle is turned at the various points in the vicinity of the magnet, they represent the direction in which the combined forces of the two poles act at these two points, hence, these lines are called magnetic lines of force.

When two magnets are suspended as in Fig. 5, the lines of force of both will be in the same direction as is indicated in Fig. 9 by the arrow heads on the curves *a a*. That this is true can be seen from Fig. 7, in which it will be seen that the arrow heads point toward the south pole and away from the north pole. As the north pole of a magnet has an attraction for the south pole, we can readily see that there is an endwise pull in the lines of force, which tends to make them contract, like rubber bands, hence, we can imagine the lines *a a* in Fig. 9 to contract and thus draw the two magnet bars together.

The repulsion of the two magnets, when the north poles are at the same end, is illustrated in Fig. 10. Here we see that the lines of force passing on the outside of the bars, as indicated by lines *a a*, are unobstructed, and can assume their natural posi-



Figs. 9 and 10. Lines of force in two bar magnets.

tion, but those that pass between the bars, along line *c*, are pressed out of position. If we assume that the lines of force make an effort to retain their position, like so many wire

springs, then we can see that the repulsion is due to the effort that the lines make to assume their natural form in the space between the bars.

Magnetic lines of force have no real existence, they simply indicate the direction in which the force acts, but if we keep this fact in mind, it helps us to understand magnetic actions, if we treat the lines of force as if they were something real. This fact will become more evident as we proceed.

Lines of force always pass from the north to the south pole through the space between these poles, and through the magnet itself, they are assumed to pass from the south to the north pole. The form of the lines of force depends upon the relative position of the north and south poles. In Fig. 9 they are curved, as

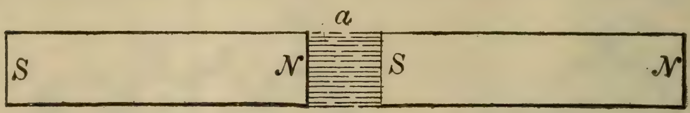


Fig. 11. Lines of force between ends of magnets.

the magnets are placed side by side, but if the bars were arranged end to end, as in Fig. 11, the lines of force would be straight, as is shown at *a*. From the north end of the right side magnet, the lines of force would pass in curved line, as in Fig. 10, to the south pole of the magnet on the left side, thus completing the magnetic chain, or circuit, as it is called.

If we take the two magnet bars of Fig. 11 and stand them on end, as in Fig. 12, and suspend a bent wire *C* in the manner shown, effects can be produced that are interesting and instructive, as they illustrate the principle upon which generators and motors act. The wire *C* should be journaled at *D D*, so as to swing with as little friction as possible. and its ends are to be connected with a battery *B*, by means of fine wires *a* and *b*; a switch being provided at *c* so as to stop the flow of current when desired.

If the switch c is opened, so that no current flows through C, the latter will not be disturbed, and if we give it a swing, it will oscillate back and forth, like a clock pendulum, and in a few seconds come to rest in the position in which it is shown. If the switch is closed, C will at once swing out of the stream of magnetic lines of force and will remain in that position as long as the current from the battery passes through it. The direction in which C

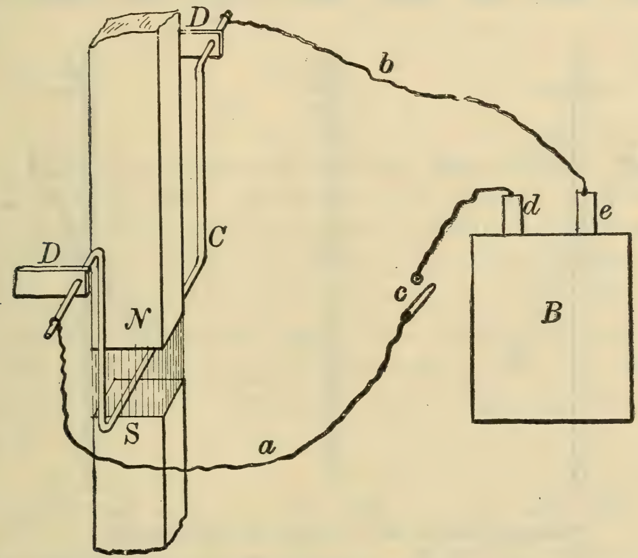


Fig. 12. Showing the principle of the electric generator.

will swing will depend upon the direction of the current through it. If with the wires a and b connected with the battery, in the manner shown, the wire C swings to the right side, then if a is connected with e, and b with d, the direction of swing will be reversed; that is, C will swing toward the left.

From this experiment we see that the magnetic lines of force can develop a repulsive force against an electric current, and that the direction of the repulsion depends upon the direction of the

electric current with respect to the direction of the lines of force. We shall now explain why this repulsion is developed, and this we can illustrate by the following experiments: —

If we arrange three wires as shown in Figs. 13, 14 and 15, so as to run north and south, the upper end being north, and place over these magnet needles *D D D*, pivoted at *e e e*, we will find that if there is no current flowing through the wire, the needle will point toward the north, or be parallel with the wire, as is

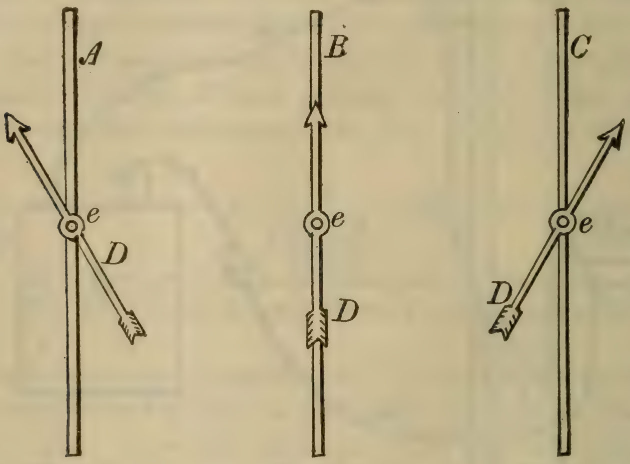


Fig. 13. Fig. 14. Fig. 15.

Showing effect of current on the needle.

shown in Fig. 14. If the current flows through the wire from south to north, the north end of the needle will swing to the right, as in Fig. 15, and if the current flows through the wire from north to south, the north end of the needle will swing toward the left, as in Fig. 13. From this we see that an electric current can repel a magnet, and that the direction in which it repels it depends upon the direction of the current.

If we stand the three wires on end, as shown in Figs. 16, 17 and 18, in which *A B C* represent the wires as seen from above, we will find out more about the relation between electric currents

and magnets. If we place four small magnet needles around each one of the wires, as shown at *a a a a*, we will find that those around the center wire, through which no current flows, will all

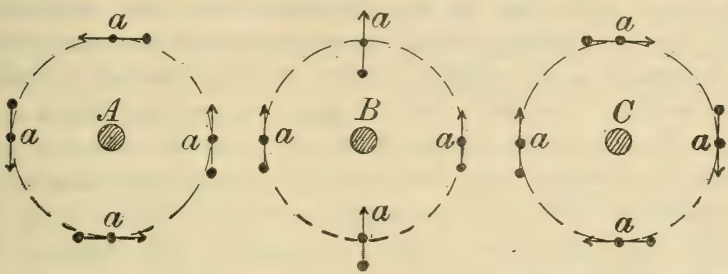
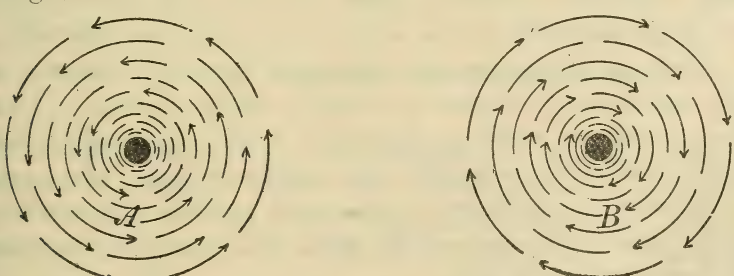


Fig. 16. Fig. 17. Fig. 18.

Wires surrounded by magnetic lines of force.

point toward the north, as shown, while those around the wire Fig. 16, through which a current flows upward, that is, away from the center of the earth, will point in a direction opposite to that in which the hands of a clock move; and in wire Fig. 18, in which the electric current flows down toward the center of the earth, the north ends of all the needles will point in the direction in which the hands of a clock move, that is, just opposite to those in Fig. 16.



Figs. 19 and 20. Directions of lines of force.

From these actions, we infer at once that when an electric current flows through a wire, the latter becomes surrounded with magnetic lines of force, as is illustrated in Figs. 19 and 20,

and that there is a fixed relation between the direction of the current and that of the lines of force. At *A*, Fig. 19, the direction of the lines of force is shown for a current moving upward, and at *B*, Fig. 20, the direction of the lines of force is that due to a current moving downward through the wire.

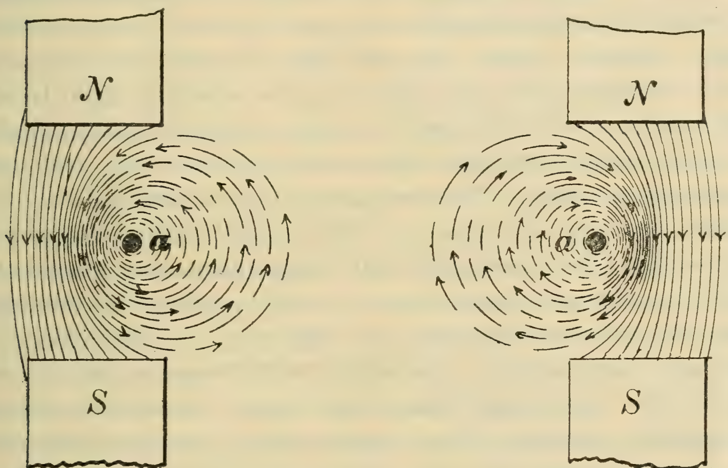
Inasmuch as an electric current flowing through a wire is surrounded by magnetic lines of force, we can say that a complete electric current consists of two parts, one the current proper, which traverses the wire, and the other the magnetic casing which envelops the wire. It is the action between the latter part of the current and the lines of force of magnets that develops the current in a generator, or the power in a motor.

With the aid of Figs. 21 and 22, we can now show how the force is developed that thrusts the wire to one side in Fig. 12. The lines of force of the magnet, which constitute what is called the magnetic field, will flow from the north pole at the top to the south pole at the bottom, as is shown in Figs. 21 and 22. If the electric current flows through the wire *C* from the back toward the front, the lines of force developed around it will have the direction shown in Fig. 21. As lines of force cannot flow in opposite directions in the same space, the lines of the field will swing over to the left side of the wire, but in doing so they will be stretched out of the straight form, and they will also push the lines surrounding the wire out of their central position. Under these conditions, which are illustrated in Fig. 21, the effort made by the field lines to straighten out, together with the effort made by the wire lines to return to the central position, will develop a thrust between the wire and the field, and thus force the former out toward the right side.

If the direction of the current through the wire is reversed so as to flow from front to back, the direction of the lines of force around the wire will be reversed, and will be as in Fig. 22. Under these conditions, the lines of force of the magnetic field will

swing over to the right side of the wire, and thus the thrust will be in the opposite direction.

Fig. 12 represents the principle of an electric motor in its simplest form, and from it we see that the force that causes the armature to rotate is developed by the repulsion between the magnetism of the field magnet and the magnetism that surrounds the wires wound upon the armature.



Figs. 21 and 22. Showing effect of magnetic field.

It is self-evident that if we undertake to force the wire *C* through the magnetic field in the opposite direction to that in which it swings, we will have to make an effort to do so; that is, if we try to move the wire from right to left in Fig. 21, or from left to right in Fig. 22, we will have to apply power. Now nature is a strict accountant and does not allow any power to be lost; therefore, all the energy we expend in moving the wire through the magnetic field must appear in some other form, and the form in which it appears is as an electric current that is generated in

the wire. If we were to remove the battery in Fig. 12 and put in its place an instrument to indicate the presence of a current in the wire, we would find that when we move the latter in the opposite direction to that in which it moves under the influence of the current, we generate a current; that is, we convert the device into a simple electric generator. If in Fig. 21, we move the wire from right to left, the direction of the current generated in the wire will be the same as that of the current which causes the wire to swing in the opposite direction, that is, from back toward the front. As it is a poor rule that does not work both ways, we would naturally infer that if moving the wire from right to left develops a current from back to front, movement in the opposite direction would develop a current from front to back; and such is actually the case. This fact can be demonstrated by Fig. 12. Suppose that in this figure we hold *C* stationary in the central position, and then pass a current through from back toward the front; this current would exert a force to swing *C* to the right side. If we release the wire, it will swing to the right and as soon as it begins to move, the current will become weaker, showing that the movement of the wire developed therein a current in the opposite direction. If we force the wire over to the left side, the current flowing through it will begin to increase as soon as the wire moves.

All the foregoing shows us that when a wire is moved through a magnetic field, a current will be generated in it if it forms part of a closed circuit, and it makes no difference whether there is a current already flowing in the wire or not. When the wire is caused to move through the magnetic field by a current flowing through it from an external source, the current developed in it will be in opposition to that which comes from the external source, and, as a consequence, the movement produces an actual reduction of the strength of current flowing through the wire. The stronger the magnetic field and the greater the velocity of the

wire, the stronger the current generated in opposition to the driving current, and, therefore, the weaker the latter. It is on this account that if a motor is allowed to run free, the faster it runs the weaker the current through it becomes, as the actual current in every case can only be the difference between the main driving current and the one developed in the wire, which latter runs in the opposite direction.

Magnetic force is measured in units that are based upon the centimeter gram second system which is too technical to be explained in a few words. Briefly stated a unit of magnetic force will exert a pull of unit mechanical force at a unit distance.

The force of magnets is measured either by the total force of the magnet, or by the force exerted by each unit of cross-section. When the measurement is based upon the total force of the magnet, the unit is called a Maxwell; thus we speak of the total flux of a magnet as so many maxwells. When the measurement is referred to the force per unit of cross-section, it is spoken of as the magnetic density, or density of magnetization, and the unit used is called a Gauss; thus we speak of a magnet as having a density of so many gausses per square centimeter, or square inch of cross-section. The density of magnetization is determined by a rule given on page 46.

The lifting capacity of a magnet can be determined by the following rule: —

TO FIND THE LIFTING CAPACITY OF A MAGNET IN POUNDS.

Multiply the area of cross-section of the magnet pole in square inches, by the square of the density of magnetization per square inch, and divide this product by 72 millions.

This rule gives the pull for one pole. For horse shoe magnets double the figures. If the object lifted is not in contact with the poles the pull will be less than rule gives.

CHAPTER II.

THE PRINCIPLES OF ELECTROMAGNETIC INDUCTION.

By electromagnetic induction, is meant the induction of electric currents by magnetic action. In the preceding chapter it has been shown that if we move a wire through a magnetic field, an electric current will be generated in it, providing its ends are joined, so as to form a closed circuit. If the ends are not joined, then there will be no current developed, because, an electric current cannot flow except in a closed circuit. When the ends of the wire are not joined, the movement through the field develops simply an electromotive force. Electromotive force is that force which causes an electric current to flow when there is a circuit in which it can flow. Electromotive force is a long-winded name and on that account it is always abbreviated into e.m.f., so that hereafter when these letters are used, it will be understood that they stand for electromotive force.

Metals and all other substances that allow electric currents to flow through them are called conductors, while glass, mica, wood, paper and many other similar forms of matter that do not allow currents to flow through them are called insulators. The difference between conductors and insulators is only one of degree, for there is no known substance that is an absolute non-conductor of electricity; that is, a perfect insulator; and there is no substance that does not resist to some extent the passage of a current — that is, there is no such thing as a perfect conductor. Some substances, like damp paper or wood, which stand midway between good conductors and good insulators, can be regarded as either one or the other, depending upon the service for which they are used. For currents of very low e.m.f., they would be in-

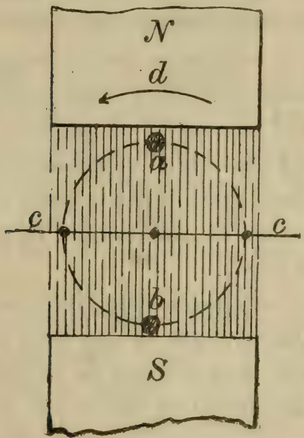
sulators, but for currents of very high e.m.f., they would be conductors.

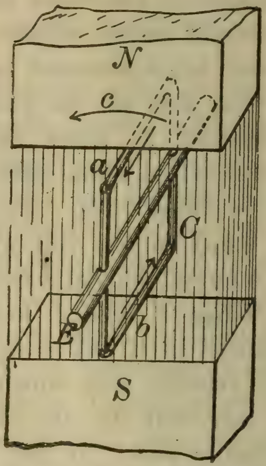
The current that will flow through any circuit when impelled by an e.m.f., will have a strength that will depend upon the amount of resistance that opposes its flow. As all conducting materials are not of the same degree of conductivity, their relative values are determined by the amount of resistance they interpose to the flow of the current. The resistance of a conductor is measured in units called ohms; the strength of current is measured in units called amperes, and the e.m.f. is measured in units called volts. The relation between these units is such that an e.m.f. of one volt will cause a current of one ampere to flow in a circuit having a resistance of one ohm.

When a wire is moved through a magnetic field, the e.m.f. induced in it will be determined by the strength of the field and the velocity with which the wire moves, and will not be affected in any way by the resistance of the circuit of which the wire forms a part. If the resistance is very great, the strength of current generated will be very low, and if the resistance is very low the current will be strong, but in either case the e.m.f. will be the same.

If movement of the wire in one direction develops an e.m.f. in a given direction through the circuit, then movement of the wire in opposite direction will reverse the direction of the e.m.f. Thus, in Fig. 23, which represents a magnetic field between the poles *N S*, if wire *a* is moved from right to left, it will have induced in it an e.m.f. that will be from back to front, and if the direction of motion of the wire is reversed, the e.m.f. will also be reversed. This will be true whether the wire is near the *N* pole or *S* pole. This being the case, it can be seen that if *a* represents the end of a wire moving in the direction of arrow *d*, and *b* the end of a wire moving in the opposite direction, the e.m.f.'s in these two wires will be in opposite directions. The

direction of the e.m.f. in a will be up from the paper toward the observer, and the direction of the e.m.f. in b will be down through the paper. If these two wires are secured to a shaft placed in the center of the field, then by the continuous rotation





Figs. 23 and 24. Illustrating the principle of the armature.

of the shaft, the two wires can be made to revolve around the circular path shown.

If these two wires are joined at the ends, as shown in Fig. 24, they will form a closed loop, and although the direction of the induced e.m.f. in the two sides will be opposite, when compared to a fixed point in space, they will be in the same direction so far as the loop is concerned; that is, both e.m.f.'s will develop currents that will flow through the wire in the same direction.

Returning to Fig. 23 it will be noticed that if the wires revolve around the circular path at a uniform velocity, their movement in the direction of line c c will not be uniform, but will be the greatest when the wires are in the position shown, and least, when they cross the line c c. In fact, when the wires cross line c c their motion in the direction of this line will be zero, for this

is the point where the direction of movement reverses. Now, the magnitude of the e.m.f. induced in the wire is proportional to the velocity in the direction of the line *c c*, hence, when the wires are crossing this line, the e.m.f. will be zero, and when they are one-quarter of a turn ahead of the line, the e.m.f. will be the highest.

In Fig. 24 we see that in side *a*, the direction of the current is toward the front, and in *b* it is the reverse; now, when *a* moves through half a turn, it will take the place of *b*, and the direction of the e.m.f. induced in it will be the same as in *b* in the figure; that is, it will be the reverse of what it is when passing in front of the pole *N*. This being the case, it is evident that each time the loop makes a half-revolution, the direction of the current generated in it reverses.

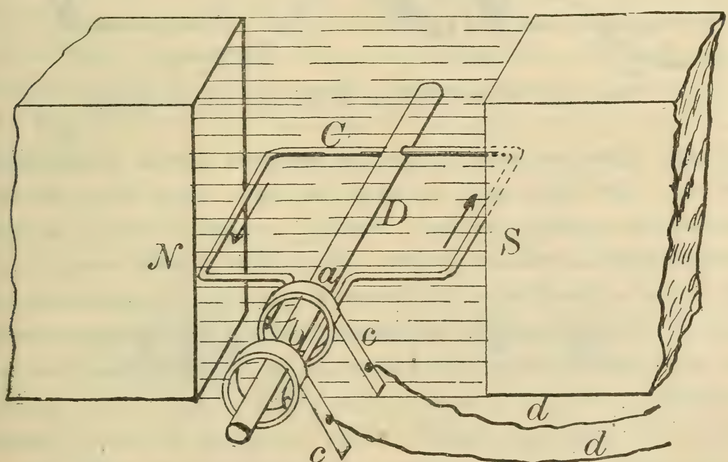


Fig. 25. Arrangement of the collector rings.

As the loop in Fig. 24 is closed, the current generated in it would be of no practical value, but if we cut the wire at one side and connect the ends with rings as shown at *a* and *b* in Fig. 25, then by means of collecting brushes *c c* we can take the cur-

rent off through the wires *d d*. This current, however, would consist of a series of impulses that would flow in opposite directions, each one starting from nothing and increasing to its greatest

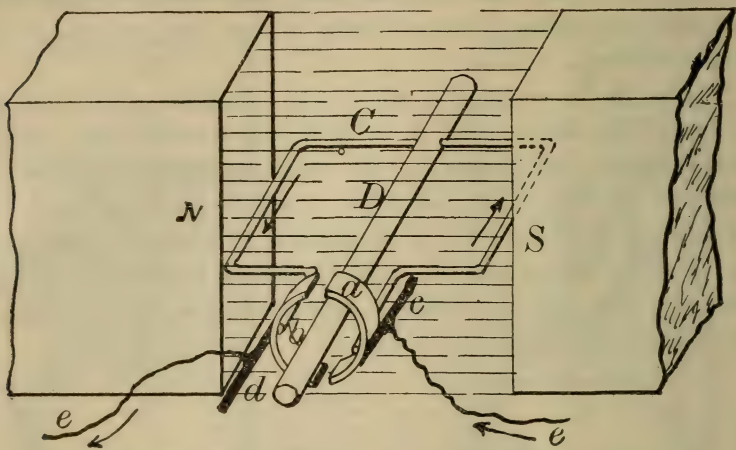


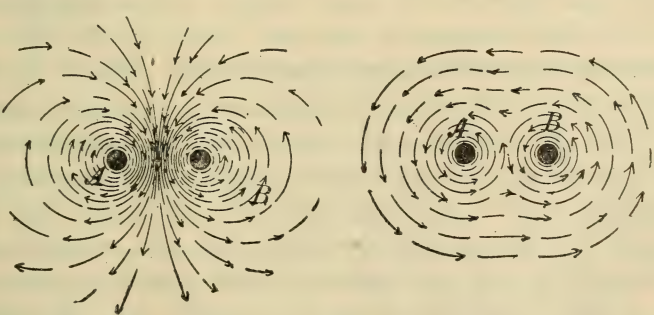
Fig. 26. Construction of simple commutator.

strength when the loop reaches the position shown in the figure, and then declining and reaching the zero value when the loop reaches the vertical position. Such a current is called an alternating current, because it flows first in one direction and then in the opposite direction. All forms of machines that generate currents by electromagnetic induction, develop alternating currents, but in the class of machines known as direct or continuous current, a rectifying device is used which rectifies the current before it reaches the external circuit. This rectifying device is called a commutator, and is illustrated in its simplest form in Fig 26. In this illustration it will be noticed that the ends of the wire, instead of being attached to two independent rings, placed side by side, are secured to two half-rings, placed opposite each other. The brushes *c d*, through which the current is taken off, are held stationary; therefore, as can be readily seen, *c* will make contact

with a during one-half of the revolution, and with b during the other half; and this will also be the case with brush d. Now, as the half-rings with which the brushes are in contact change at each half revolution, it follows that by properly setting the brushes, they can be made to pass from one-half ring to the other at the very instant when the direction of the current in the loop reverses, so that through each brush there will be a succession of current impulses, but all in the same direction.

The device shown in Fig. 25 is a perfect alternating current generator, and that shown in Fig. 26 is a perfect direct current generator. In both cases, however, the e.m.f. induced is so low as to be of no practical value. To obtain serviceable machines, capable of developing the e.m.f. and current strength required in practice, it is necessary to provide very strong magnetic fields and to rotate in these a large number of loops of wire. In order that the operation of such machines may be understood, we will first show how the powerful magnetic fields are obtained.

In Fig. 27 two wires are shown as seen from the end, these being marked A and B. The lines of force surrounding them are



Figs. 27 and 28. Effects of direction of current in wires.

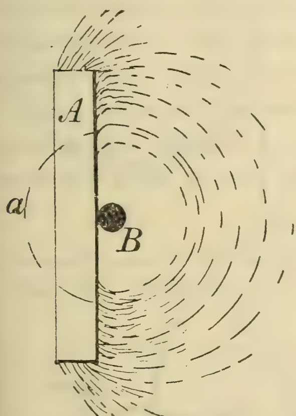
in directions that correspond to opposite directions of current in the wires. In wire A, the current flows away from the observer. As can be seen, the lines of force of both wires have to crowd into

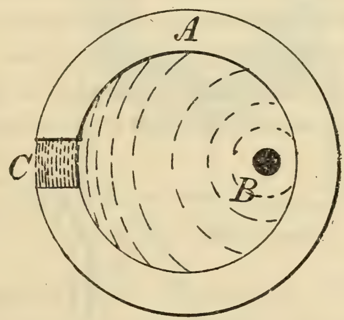
the space between the wires, for on the outside of A the two sets of lines would meet each other head on, and this would also be the case on the right side of wire B. This crowding of the lines of force into the space between the wires causes them to distort from their natural position and instead of being central with the wires, are eccentric to them. If we take a long wire through which a current is flowing and bend it into a loop, we will see that if the current flows out through one side, it will return through the other side, so that in the two sides of the loop the current will flow in opposite directions. This being the case, Fig. 27 can be regarded as showing the two sides of such a loop, and from it we find that the effect of such a loop is to concentrate within its interior nearly all the lines of force that surround the wire.

In Fig. 28 the two wires A and B are surrounded with lines of force that correspond to the same direction of current. In this case it will be noticed that in the space between the wires the lines of force flow in opposite directions; hence, only a few of the lines will follow this path, simply that number surrounding each wire that can traverse the space without encroaching upon the path of the lines belonging to the other wire. If the two wires are very near to each other, practically all the lines of force of both wires will join forces, so to speak, and pass around the two wires. Now, if we wind a wire into a coil of many turns, the direction of the current in the several turns will be the same, so that the lines of force of all the turns will combine into one large stream and circulate around the entire coil side, no matter how many turns of wire it may contain. From this it can be seen that if we have a current of, say, ten amperes, we can make it produce just as powerful magnetic effect as a current of one thousand amperes, by simply increasing the number of turns of wire in the coil. A current of ten amperes passing through a coil of wire containing one hundred turns, will have the same magnet-

ism in effect, as a current of one hundred amperes passing through a coil of ten turns, or as a current of one thousand amperes passing through a coil of a single turn.

If we place at the side of a wire through which an electric current is flowing a piece of iron, as is shown in Fig. 29, the effect will be that the lines of force will no longer flow in circular paths, as indicated by the circle *a*, but will be deflected in the manner illustrated, by the presence of the iron. If, instead of





Figs. 29 and 30. Lines of force through wires and magnets.

the straight iron bar, we substitute a ring of iron, as in Fig. 30, nearly all the lines of force will be concentrated in the metal, and the magnetic field in the space *C*, between the ends of the ring, will be vastly greater than at any other point. The explanation of these actions is that all forms of matter oppose the development of magnetic force, but some offer greater resistance than others. Iron, steel, nickel, and one or two other metals, offer less resistance to the magnetic lines of force than air, and are said to have a higher magnetic permeability. Nickel is only a slight improvement on air, but steel and iron are far superior, iron being of about two to three times the permeability of hard-

ened steel, and about one thousand times the permeability of air, when magnetized to the density ordinarily used in practice. The iron in Figs. 29 and 30, therefore, becomes the path of the lines of force, because it interposes a much lower resistance. Owing to this difference in the resistance of iron and air, it is possible to make an iron magnet core of any desired form, and to concentrate within it nearly all the lines of force developed by the current flowing through the wire wound upon it. The presence of the iron not only serves to concentrate the magnetism in it, but as it reduces the resistance opposing the development of the magnetism, it enables the field to be made vastly stronger than it could be with air alone, say a thousand times as great.

If we make a magnet in the form of Fig. 31, with a coil of wire around the part *B*, practically all the lines of force will flow to

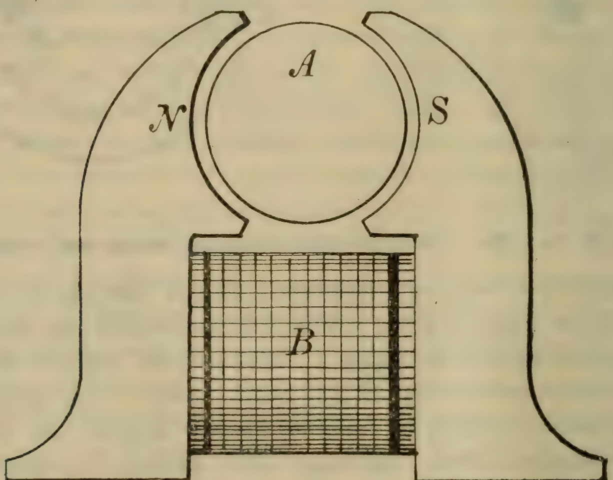


Fig. 31. Principle of construction of bipolar machine.

the poles *N S*, and will pass through the air space between them. If this air space is nearly filled with a cylindrical mass of iron, *A*, the strength of the magnet will be increased, for, by doing this,

we replace air which is a poor magnetic conductor, by iron which is a far superior conductor. Electric motors and generators are made with a cylindrical mass of iron at *A*, which is the armature

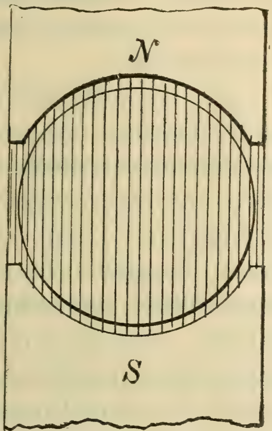


Fig. 32. Solid core.

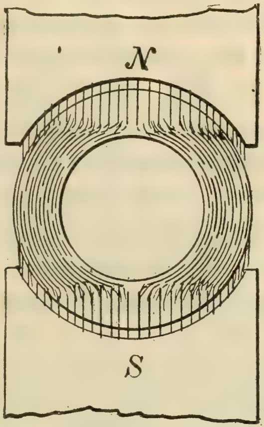


Fig. 33. Ring core.

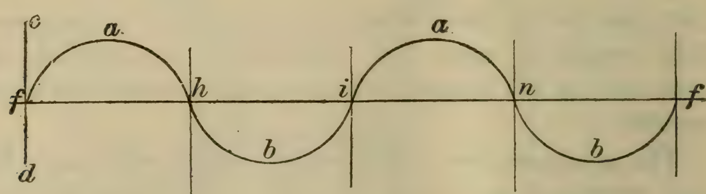
core, and the air space between it and the faces of the poles of the field magnet is made just sufficient to accommodate the wire coils, and by this means the field strength is increased as much as possible.

The armature cores are sometimes made solid, as in Fig. 32, and sometimes as a ring, as in Fig. 33. When they are solid, the lines of force cross through them in straight lines, see Fig. 32; and when they are ring form, the lines follow the ring and do not penetrate the interior space.

If the single loop of Fig. 24 is replaced by a coil containing many turns of wire, the e.m.f. induced in it will be increased in proportion with the number of turns of wire in the coil, so that by using such a coil in a field such as shown in Fig. 31, a high e.m.f. can be obtained. This e.m.f., however, would be alternating, and if the current were rectified by means of a commu-

tator, it would not be of uniform strength, but would fluctuate from a maximum value to zero. Just how the current would fluctuate and how the construction can be changed so as to get rid of the fluctuation, we can explain by presenting a diagram that illustrates the alternating current as it flows in the armature coil, and the rectified current as it leaves the commutator.

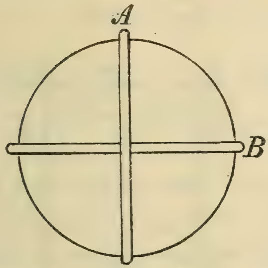
In Fig. 34, let the distance *f h*, *h i*, *i n*, along the line *f f* represent half-revolutions of the coil, and let distances measured on the vertical line *c d* represent the strength of current, distances above *f* being current flowing in one direction, and distances below *f* being for current flowing in the opposite direction. Let us consider the instant when the coil is passing the point where the e.m.f. induced is zero; then this instant will be represented by the point *f*, at the left of the diagram, and the curve *a* will start from this point; as at that instant, the current which it represents has no value. As the coil rotates, the current begins to grow, and this fact we indicate by causing curve *a* to gradually



Figs. 34 and 35. Illustrating flow of alternating current.

rise above the horizontal line. At the quarter turn, the current reaches its greatest strength, thus this forms the highest point of curve *a*, and is midway between *f* and *h*. From this point

onward, the current declines and becomes zero, when the rotation of the coil has reached one-half of a revolution, which is represented by the point h. In the next half-revolution, the current



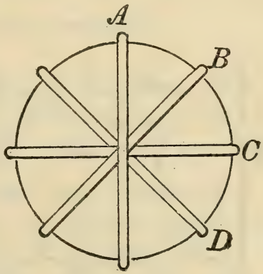


Fig. 36. Two coils on armature. Fig. 37. Four coils on armature.

flows in the reverse direction, but has the same maximum strength and increases and decreases at the same rate; therefore, the curve b, drawn below the horizontal line, represents the reverse current; and point i corresponds to one complete revolution, so that beyond i the curves a and b are repeated in systematic order.

Now, if we provide a commutator to rectify this current, all we can accomplish is to turn curve b upside down and transfer it to the upper side of the horizontal line, as in Fig. 35; but, as will be seen, all we accomplish by this act is to obtain a current that flows always in the same direction, but at each half-revolution it drops down to a zero value.

If we wind two coils upon the armature, placing them at right angles with each other, as is indicated by A and B in Fig. 36, then if the currents of these two coils are rectified, they will bear the relation toward each other shown at the upper line in Fig. 38, the a a curves in solid lines representing the current from the A coil, and the b b curves in broken lines, representing the current from the B coil. As will be seen, when one of these currents is zero, the other is at its greatest value, so that if we run both into

the same circuit, the lowest value of the combined current would be equal to the maximum of either one of the single currents, and the maximum value would be equal to the sum of the two currents when the coils are on the eighths of the revolution.

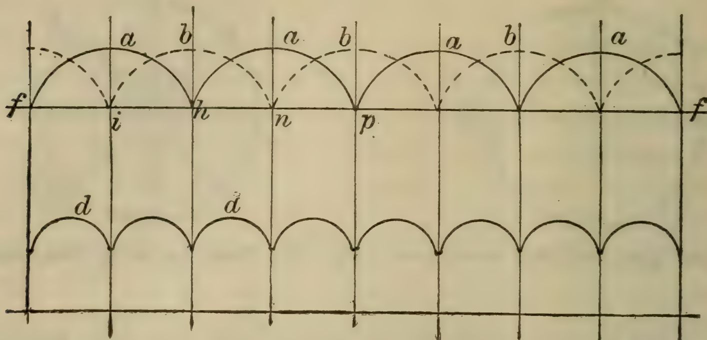


Fig. 38. Showing effect of larger number of coils.

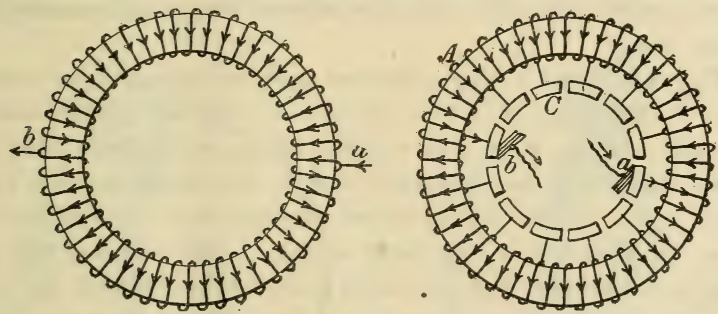
This resulting current is shown on the lower line in Fig. 38 by the curve *d d*. From this curve we see that the number of fluctuations in the current has been doubled, but the variation in the strength is greatly reduced. If we wound four coils upon the armature, as indicated by *A B C D*, in Fig. 37, the number of undulations in the combined current would be again doubled, but the fluctuation would be very much less. If the number of coils is increased to twenty-five or thirty, the fluctuations in the current become so small as to be hardly worth noticing.

With coils such as shown in Fig. 26, a separate commutator would have to be provided for each coil, and this would render the machine very complicated, if the number of coils were even six or eight; hence, in actual machines, the winding of the coils is modified so as to be able to use a single commutator for any number of coils. This construction will be explained in the next chapter.

CHAPTER III.

TWO POLE GENERATORS AND MOTORS.

The simplest type of armature winding is that used with ring cores, and is illustrated in Fig. 39. As will be seen, it is simply a continuous winding all the way around the circle, the end of the last turn of wire being connected with the beginning of the first turn, so as to form an endless coil. If wires are attached at *a* and *b*, and a current is passed through, it will divide into two halves, one part flowing through the wire above *a b*, and the other part through the wire below *a b*. In the upper half of the wire, the direction of the current in the front sides of the turns will be toward the center of the ring, as is indicated by the arrow heads, and in the lower half it will be away from the center. If, in-



Figs. 39 and 40. Windings on ring armatures.

stead of attaching wires at *a* and *b* we place stationary springs, so as to press against the wire, then we could revolve the ring, and still the current would enter and leave the wire at the same points. Small armatures are often made in this way, but for regular

machines it is more desirable to provide a commutator as shown in Fig. 40 at *C*, and then the several segments can be connected with the wire at regular intervals. In the figure, the commutator is provided with twelve segments, and these connect with the armature wire at every fourth turn, so that the wire is divided into twelve coils of four turns each.

The only difference between this diagram and a regular generator armature of the ring type, is that it shows the wire coils spread out with a considerable space between them, and only in one layer, while in the actual machine, the wire is wound close together and generally, in several layers; but no matter how many layers there may be, or how many turns in a coil, the principle of winding is the same.

We have shown the ring winding first, because it is so simple that it can be understood with the most superficial explanation. The drum winding, which is used to a much greater extent, is the same in principle as the ring, but owing to the fact that the coils cross each other at the ends, it appears to be decidedly different. By the aid of Figs. 41 to 44, the drum winding can be made perfectly clear.

Fig. 41 shows a ring armature core with a single coil wound upon it; and Fig. 42 shows a drum core, with a single coil wound upon it. In the ring, only one side of the coil appears upon the outer surface of the armature, but in the drum, as there is no open space for the coil to thread through, both sides of the coil must be placed upon the outer surface. The side *B* of the coil may be called the live side, as it is the one from which the ends project, and the lower side *c*, may be called the dead side. Since only the live side of the coil has ends to be connected, it can be readily seen that if in the drum winding we leave spaces between the live sides for the dead sides, and then connect the ends of the live sides by jumping over the dead side between them, that we will have the same order of connection as in the ring winding.

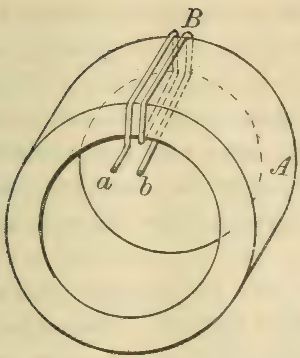


Fig. 41. Ring armature core.

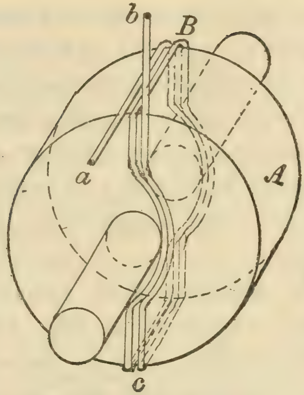
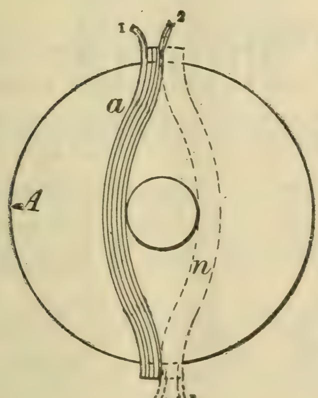
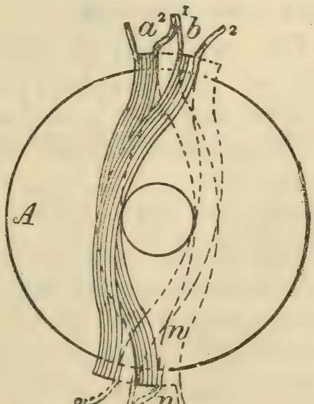


Fig. 42. Drum armature core.

The dead side of each coil adjoins the live side of a coil that is, in reality, half a circumference away from it; thus, in Fig. 43, the live side of coil *a* is at the top and the dead side is at the bottom; while the live side of coil *n* is at the bottom and the dead side is at the top. The live sides of these two coils are on opposite sides of the armature, so that the coil side to the right of *a* is simply





Figs. 43 and 44. Windings of drum armature.

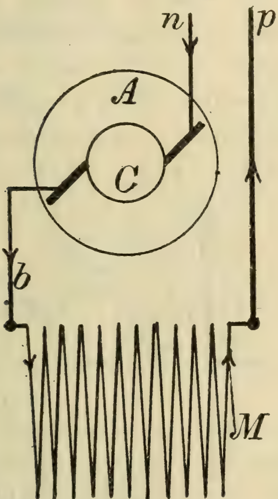
the dead side of a coil whose live side is on the other side of the armature. In Fig. 44 the two coils *a* and *b* are adjoining coils, for the coil side between them is the dead side of coil *n*. To connect the armature, therefore, we join end 2 of coil *a* with end 1 of coil *b*, and the end 2 of coil *b* would jump over a dead side and connect with end 1 of coil *c*. Coil *c*, however, would appear to be two coils ahead of *b*, just as *b* appears to be two coils ahead of *a*.

In winding drum armatures, the coils are generally placed in pairs, as shown in Fig. 43 and also in Fig. 44. The object of this is simply to make the ends of the armature look more even. A drum armature can be wound out of a continuous wire, by simply making a loop to take the place of the ends 1 and 2, and then skipping a space, as shown by coils *a* and *b* in Fig. 44. After the armature is half covered, there will be spaces left between the coils, these spaces being of the width of a coil; we then proceed to fill up the vacant spaces, and when they are all filled, the last coil put in will be the proper position to connect with the first one wound. A little practice with a piece of twine and a wooden cylinder, will enable any one to find out in short order how to wind drum armatures.

The two types of winding just explained, are those used with two pole machines, motors as well as generators. It may be added that there is no difference, electrically, between a motor and a generator, and any machine can be used for either service. Motors, however, are somewhat modified in design so as to make them more suited to the work they have to perform. The modification consists mainly in protecting the parts liable to be injured by objects falling upon them.

The general arrangement of the field and armature in a two pole machine is shown in Fig. 31. The design can be changed in a vast number of ways, but it will always be two-pole, or bipolar, as it is called, if only two poles are presented to the armature.

Generators and motors are arranged so that the current that magnetizes the field may be the whole current that flows in the circuit, or only a part of it. When the whole current passes through the field magnetizing coils, the machine is said to be of the series type; this name being given because the armature wire and the field coils are connected in series, so that the current first passes through one and then through the other. If the field coils are traversed by only a portion of the current, the machine



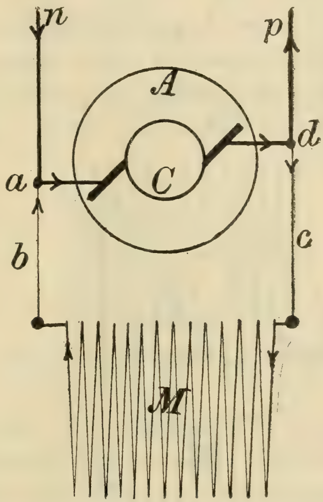


Fig. 45. Showing series connection. Fig. 46. Showing shunt connection.

is said to be of the shunt type, owing to the fact that the field is supplied with a current that is shunted from the main circuit. Generators and motors are also arranged so that there are two sets of field coils and one is traversed by the whole current, and the other by a portion thereof. The best way to understand these different types of connection is by means of simple diagrams that show the wire coils of the field and the outline of the armature. Such diagrams are presented in Figs. 45 to 50. Fig. 45

represents the series connection. *A* being the armature, *C* the commutator, and *M* the field coil. The direction of the current is indicated by the arrow heads. Fig. 46 is the shunt connection, and the arrow heads show the direction of the currents in the case of a generator. As will be seen, at *d* the field current branches off from the main line and returns to it at *a*, after having passed through the field coil. Fig. 47 shows the type in which the field is magnetized by two sets of coils, one being in series with the main circuit and the other in shunt. As will be noticed, all the armature current passing out through wire *d*, goes through coil *F*, except the portion that is shunted at *c*, into the shunt coil *M*. This type of winding is called compound, being a combination of the series and shunt. When the shunt coil is connected as in

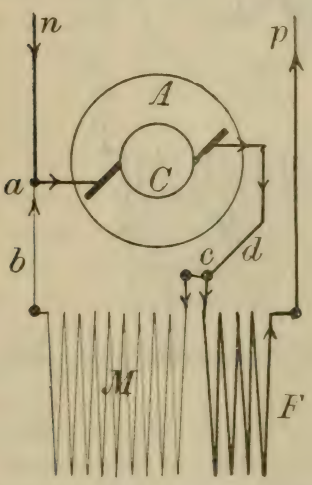


Fig. 47. Field magnetized by two sets of coils.

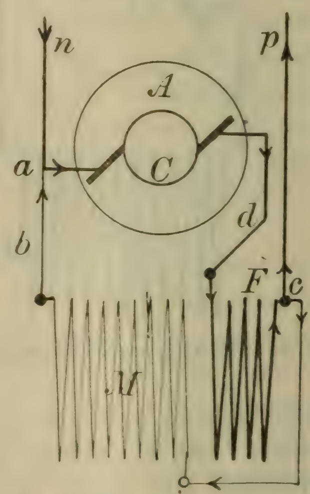
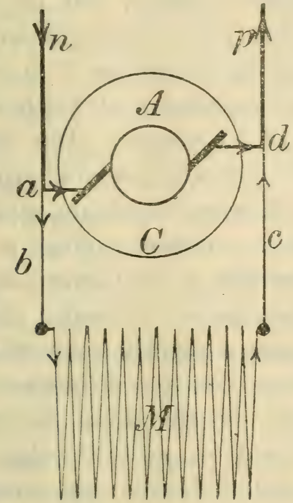


Fig. 48. Illustrating long shunt.

Fig. 47, it is called a short shunt, and when as in Fig. 48, it is a long shunt. In the first case, the coil *M* shunts the armature only, and in the second, it shunts the coil *F* also.

Figs. 49 and 50 show the shunt and compound types for motors, and as will be noticed, the only difference between them and the generator diagrams, is that the direction of the current



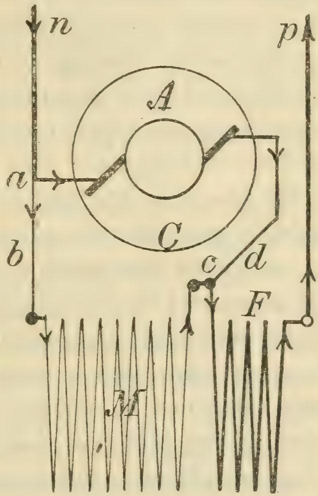


Fig. 49. Shunt type of motor. Fig. 50. Compound type of motor.

in the shunt coils is not the same. This difference in direction is due to the fact that in the generator the armature generates the current that passes through coil *M;* hence, at point *d*, the current flows up to the main line and down to the field coil. In the motor, the current comes from an external source through main *n*, and thus passes from *a* to the armature, and also to the field coil, thus traversing the latter in the opposite direction. In the series coil *F*, the direction of the current is the same in both machines.

Generators are made so as to keep the strength of the current constant, and allow the voltage to vary as the demands of the service may require; or they may be wound so as to keep the voltage constant and allow the current strength to vary. Machines

of the first class are called constant current, and are used principally for arc lighting. Machines of the second class are called constant potential and are the kind used for incandescent lighting, for electric railways and for the operation of motors of every description. For constant current generators the series winding is used in connection with some kind of regulating device that prevents the current strength from varying more than the small fraction of an ampere. The shunt and compound windings are used for constant potential generators. If the armature wire had no resistance, the shunt winding would enable a generator to maintain a constant voltage at its terminals, no matter how much the strength of the current might vary; but armatures without resistance cannot be made; therefore, a shunt-wound machine will develop a slightly lower voltage with full current than with a weak one, but the difference will not be more than three to five per cent. By the aid of the compound winding, the generator can be made so as to develop the same voltage with light or full load, and if desired, the voltage can be made to increase as the current increases. If a compound generator is so proportioned that the voltage is the same for weak and strong currents, it is said to be evenly-compounded, and if the voltage increases as the current increases, it is said to be over-compounded. If the voltage is five per cent higher, with full load than with no load, the generator is said to be over-compounded five per cent, and if the increase is ten per cent, it is said to be over-compounded ten per cent.

The way in which a compound generator increases the voltage can be readily understood from an examination of Fig. 47. The current that passes through the shunt coil M, is practically one of the same strength at all times; therefore, the magnetizing effect of this coil does not change. Through coil F the whole current passes, hence, the magnetizing effect of this coil increases as the current strength increases. Now the total field

magnetism is that due to the combined action of the two coils, so that as the action of F increases, the strength of the field increases. If F has only a few turns of wire, it will only help slightly to magnetize the field; therefore, its increased effect, due to increase in current, will not be very noticeable; but if F has many turns, it will develop a large proportion of the field magnetism, and, under this condition, the change in current strength will make a decided change in the strength of the field, and thus in the voltage, for the voltage is directly proportional to the strength of the field.

In motors, the coil F can be connected so as to act with coil M, or against it. If both coils act together, the motor is compound-wound; and if F acts against M, the motor is differentially-wound. A compound-wound motor will slow down more with a heavy load than a simple shunt machine, but it will carry the load with a smaller current, and, on this account, this winding is commonly used for elevator motors. A differential motor will hold up the speed better with a heavy load than a simple shunt machine, but it will take a correspondingly larger current to do the work. The differential winding is not used to any great extent, except in cases where it is desired to obtain as uniform a velocity as possible.

In explaining the principles of armature winding, it was shown that the commutator brushes must make contact with the commutator on the sides, that is, that in Fig. 51, they would be placed on the diameter $n\ n$. In actual machines, they are either ahead of this line, as in Fig. 52, or back of it, as in Fig. 53. The first position is that of the generator and the second that of the motor. The reason why the brushes have to be set ahead of line $n\ n$ in a generator, and back of the line in a motor, is that the armature current develops a magnetization of its own, and this reacts upon the magnetism of the field so as to twist the lines of force out of their true path. If we look at Fig. 39, we can see

that the direction of the current through the wires is such that the magnetizing effect produced upon the armature core is the same as it would be if the wire were wound in the way indicated by the vertical lines in Fig. 51. Now this current will develop a magnetization in the direction of line *n n*; that is, at right angles to the field magnetism. These two magnetic forces of the armature and the field, engage in a tug of war, and the result is that the actual magnetization that acts upon the armature wire is the combined effect of the two. If the strength of the field magnetism

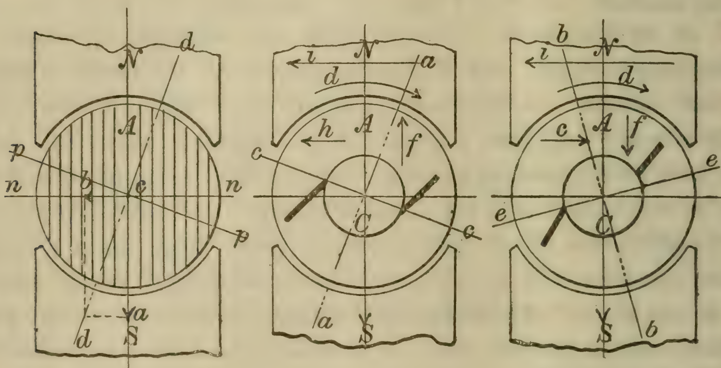


Fig. 51. Fig. 52. Fig. 53.

Showing proper position of brushes.

is proportional to line *c a*, and the strength of the armature magnetization is proportional to line *c b*, then the actual magnetization will be equal to line *c d*, and in the direction *d d*. In Fig. 52, which represents a generator, if the current in the field coils passes over the front side in the direction of arrow *i*, and the armature revolves in the direction of arrow *d*, then the armature current will be in the direction of arrow *f* and the armature magnetization will be in the direction of arrow *h*. The field magnetization will be from *N* to *S*, therefore, the resulting magnetization will be in the direction of line *a a*. Now the proper position for

the brushes is on a line at right angles to the direction of the field, hence, they must rest upon line *c c*. If the machine is a motor, the only change effected will be that the direction of the armature current will be reversed, so that arrow *f* will point downward instead of upward, and the magnetism of the armature will be directed to the right as shown by arrow *c*. Under these conditions, the actual direction of the field magnetism will be that of line *b b*, and upon line *e e*, at right angles to this the brushes must be set.

WIRING TABLE FOR 110-VOLT, 16 CANDLE POWER LAMPS.

(Size of Wire in B. & S. Gauge.)

No. of Lamps.	DISTANCE IN FEET TO CENTER OF DISTRIBUTION.																
	20	25	30	35	40	45	50	60	70	80	90	100	120	140	160	180	200
2	20	19	19	19	19	19	19	19	18	18	17	16	16	15	15	14	13
3	19	19	19	19	19	18	18	17	16	16	15	15	14	13	13	12	12
4	19	19	19	18	18	17	16	16	15	14	14	13	13	12	11	11	10
5	19	19	18	17	16	16	16	15	14	14	13	13	12	11	11	10	10
6	19	18	17	16	16	15	15	14	14	13	13	12	11	11	10	10	9
7	18	17	16	16	15	15	14	14	13	12	12	11	11	10	9	9	8
8	18	17	16	15	15	14	14	13	12	12	11	11	10	9	9	8	8
9	17	16	15	15	14	14	13	12	12	11	11	10	9	9	8	8	7
10	17	16	15	14	14	13	13	12	11	11	10	10	9	8	8	7	7
12	16	15	14	14	13	13	12	11	11	10	10	9	8	8	7	7	6
14	15	14	13	13	12	12	11	10	10	9	9	8	7	7	6	6	5
16	15	14	13	12	12	11	11	10	9	9	8	8	7	6	6	5	5
18	14	13	12	11	11	10	10	9	9	8	8	7	6	6	5	5	4
20	14	13	12	11	11	10	10	9	8	8	7	7	6	5	5	4	4
25	13	12	11	10	10	9	9	8	7	7	6	6	5	4	4	3	3
30	12	11	10	10	9	8	8	7	6	6	5	5	4	3	3	3	2
35	11	10	10	9	8	8	7	7	6	5	5	4	4	3	2	2	1
40	11	10	9	8	8	7	7	6	5	5	4	4	3	2	1	1	1
45	10	9	8	8	7	7	6	5	5	4	4	3	2	2	1	1	0
50	9	9	8	7	7	6	6	5	4	4	3	3	2	1	1	0	0
60	8	8	7	7	6	6	5	4	3	3	2	2	1	1	0		
70	7	7	7	6	5	5	4	4	3	2	2	1	1	0	0		
80	6	6	6	5	5	4	4	3	2	2	1	1	0	0			
90	6	6	5	5	4	4	3	2	2	1	1	0	0				
100	5	5	5	4	4	3	3	2	1	1	0	0					

CHAPTER IV.

MULTIPOLAR MACHINES.

The only difference between a bipolar and multipolar machine is, that the latter has two poles, and the former has two or more pairs of poles. In consequence of this difference in the number

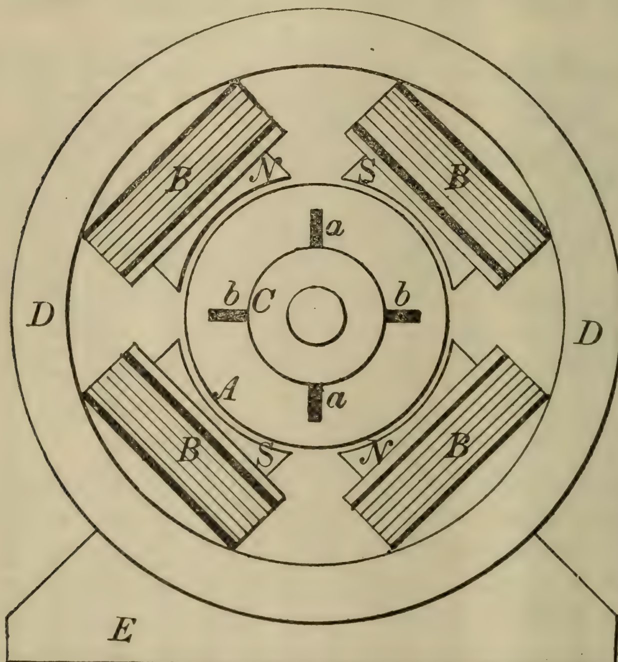


Fig. 54. Showing four-pole machine.

of poles, the armature winding has to be slightly modified, as will be presently explained. Fig. 54 illustrates a four-pole machine

and, as will be noticed, the N and S poles alternate around the circle. This arrangement is followed, no matter what the number of poles may be.

The advantage of the multipolar construction is that it increases the capacity of the machine for a given size and weight. Figs. 55 to 57 illustrate the gain effected in weight. The first figure shows a two-pole machine, the second a four-pole and the third an eight-pole, the three being of the same capacity. The poles of the second machine are half as wide as those of the first, as there are twice as many. The other parts are reduced in like proportion. In Fig. 57, the poles are one-quarter as wide as in

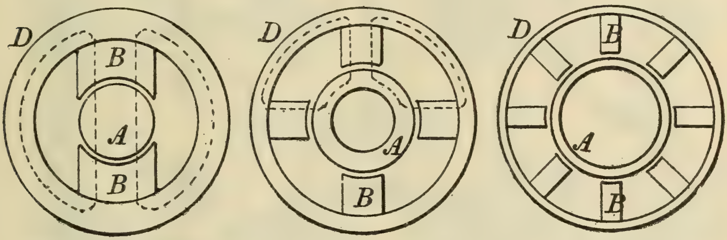


Fig. 55. Fig. 56. Fig. 57.

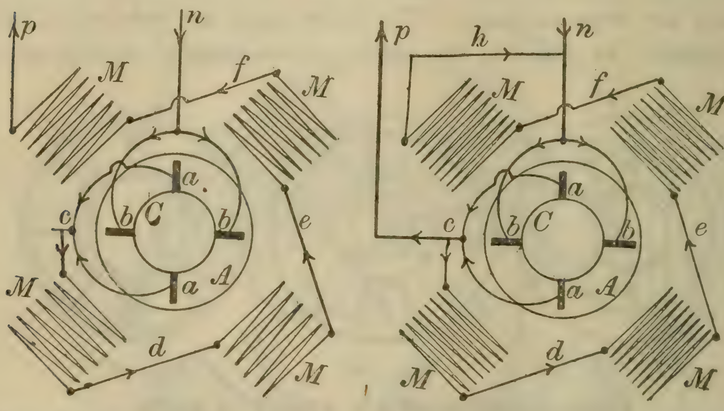
Effect of increasing the number of poles.

Fig. 55, as there are four times as many. On account of the reduction in the width of the poles, the armatures can be increased in diameter as the number of poles is increased, without increasing the outside dimensions of the machine, so that in reality, Fig. 56 is somewhat more powerful than Fig. 55, and Fig. 57 is still more powerful.

The fields of multipolar machines are wound the same as those of the bipolar; that is, as series, shunt or compound. Figs. 58 to 60 show the three types of winding for a four-pole machine and Fig. 61 is a diagram of compound winding for an eight-pole generator. The number of commutator brushes used is equal to the number of poles, although with one type of armature

winding, two brushes are sufficient, no matter how many poles the machines may have. In practice, however, even with this winding, the number of brushes is generally made equal to the number of poles.

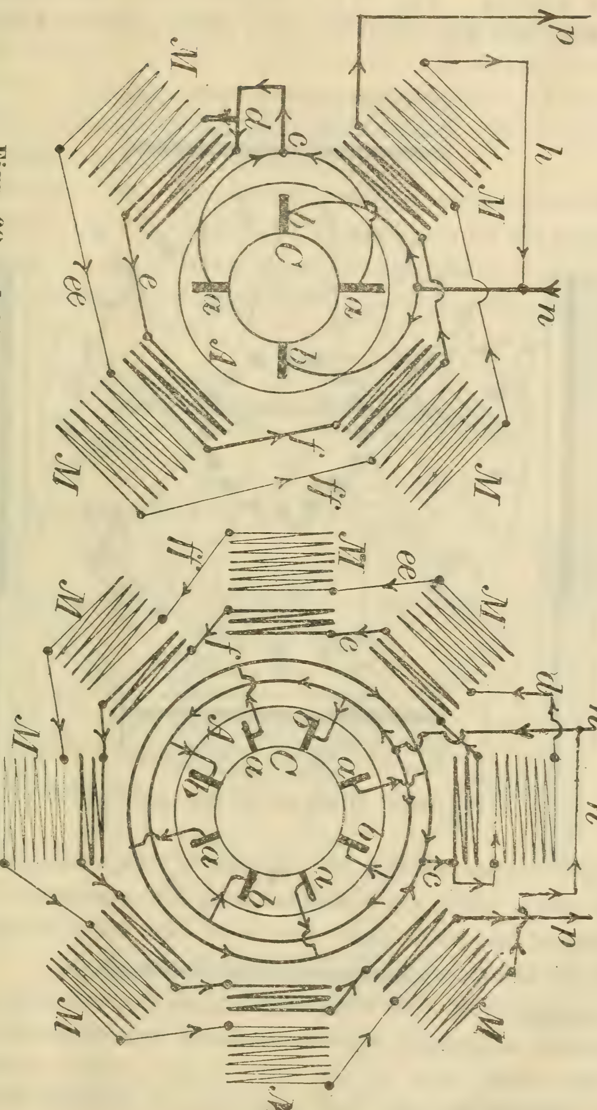
With a four-pole machine the brushes can be connected in a simple manner, as shown in Figs. 58 to 60, but with a greater number of poles, two rings are generally provided, to which the brushes are connected in the manner shown in Fig. 62.



Figs. 58 and 59. Connection of brushes on four-pole machine.

Looking at Fig. 54, it can be seen that if the current flows up from the paper, under the *N* poles, it will flow down through the paper, under the *S* poles; hence, the armature coils in a four-pole machine must span only one-quarter of the circumference, and not one-half, as in the two-pole armature. For a six-pole armature, the coils must span one-sixth of the circumference, and for an eight-pole, one-eighth, and so on, for any higher number of poles.

There are two types of winding for multipolar armatures, one being called the lap, or parallel winding, and the other the wave



Figs. 60 and 61. Connection of brushes in multipolar machines.

or series winding. Fig. 62 is a diagrammatic illustration of the lap winding, and Fig. 63 of the wave winding, both for four poles.

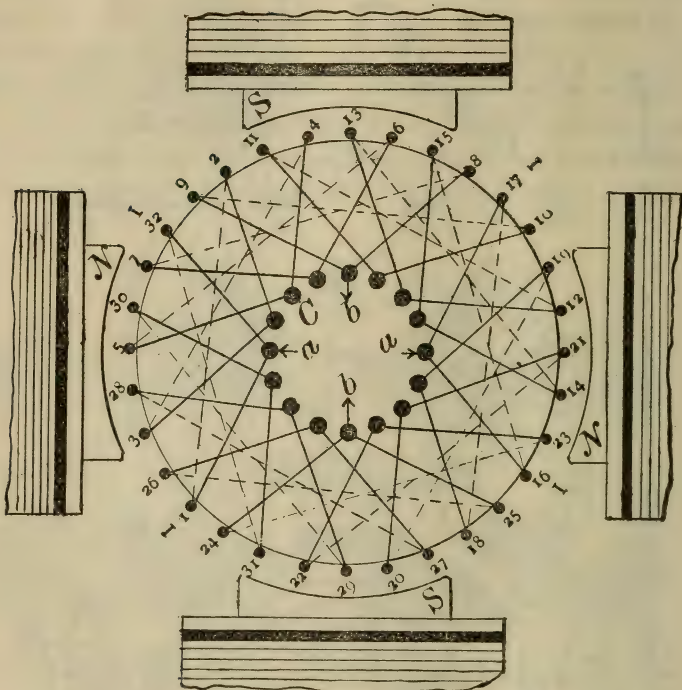


Fig. 62. Diagram of lap winding.

The small circles around the outside of the armature represent bars or wires, which are connected with the commutator segments by means of the solid lines, and with each other at the opposite side of the armature, by means of the broken lines.

If we start from coil side, or bar 1 on the left, and follow the connections as guided by the numbers, we will finally reach 32, and thus come back to left side brush *a*, which is the starting point. As will be seen, bar 1 connects at the back of armature,

with bar 2, and then over the front, the connection runs in the backward direction, to bar 3; thence, forward again, at the back end, to bar 4, and again backward over the front, to bar 5. The connections, therefore, lap over each other and it is on this account, that it is called a lap winding.

Fig. 63 shows the wave winding, and it will be noticed that if we start from bar 1 at the top, we advance around the right to bar 2, and then we go further ahead to bar 3, and in like manner advance to bar 4, the connections in every case advancing in the

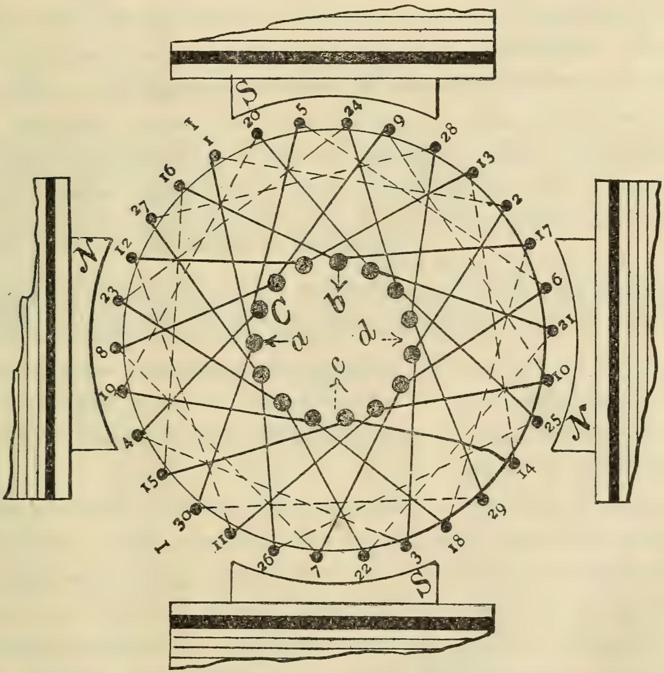


Fig. 63. Diagram of wave winding.

same direction around the circle. It will be further noticed that the connections run zig-zag from side to side of the armature core

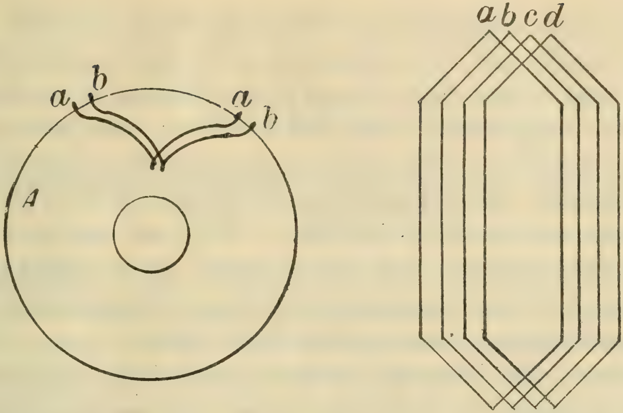
as they advance, thus forming a wave-like path for the current, and it is on this account that this style of connection is called wave winding.

With the lap winding, the brushes *a a* are connected with each other, and so are the *b b* brushes. In the wave winding, two brushes set one-quarter of the circle from each other, will take the current off properly as indicated by *a* and *b* in Fig. 63, but four brushes can also be used.

In Fig. 54, the brushes are shown midway between the poles, while in Figs. 62 and 63, they are opposite the poles. This difference in position is due to the fact that in the last two named figures, the connections between the armature coils and the commutator segments do not run in radial lines from either side, but one connection bends backward and the other forward. In actual machines, the connections are run as in these diagrams, and in some cases, one of the sides runs in a radial direction; therefore, in some generators, the brushes are opposite the poles, and in others they are between them.

Diagrams 62 and 63 show coils of a single turn, but by regarding the broken lines as representing the position of the end of the coil at front as well as the back of the armature, and the solid lines as simply the ends of the wire that connect with the commutator segments, they become accurate representations of coils of any number of turns.

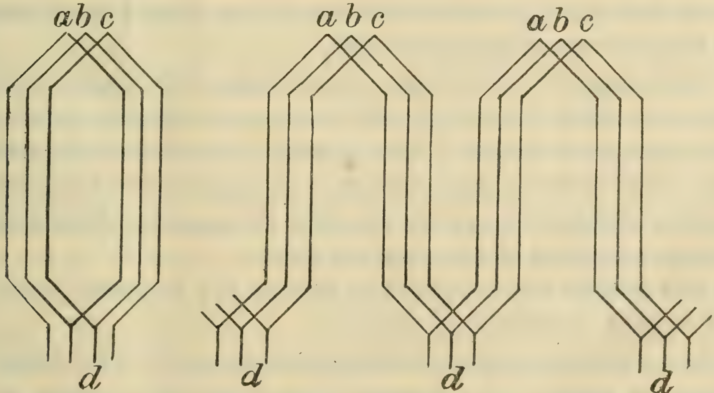
The coils of multipolar armatures are made on forms, and in the finished state are placed upon the armature core. Some coils are so formed as to bend down over the ends of the armature, and are then given the form at the ends, shown in Fig. 64, so they may fit into each other. In some machines, the coils do not bend down over the ends of the armature, but run out parallel with the shaft. Armatures so wound are sometimes said to have a barrel winding, and the coils, if laid out upon a flat surface, would present the appearance of Fig. 65; that is, if they con-



Figs. 64 and 65. Armature windings.

tained more than one turn. If of the single-turn type, they would look like Fig. 66, if for a lap winding; and like Fig. 67, if for a wave winding, the ends *d d* being joined and then connected with the commutator segments.

In connecting the field coils of multipolar machines, it is necessary to be careful not to make mistakes, so that some of the



Figs. 66 and 67. Drum and barrel windings.

coils will act to magnetize the field in the wrong direction. By studying Fig. 27 and the explanation of it, the direction of the magnetic lines of force with respect to the direction of the current through the magnetizing coils, can be clearly understood, and then there will be no difficulty in determining the proper way in which to connect the coil ends, for all we have to do is to make the connections such that if one pole is *N* the one next to it is *S*. With two-pole machines, it is also necessary to be careful not to connect the field coils improperly; that is, if there is more than one coil, and in most machines this is the case.

The current that energizes a magnet is called the magnetizing force and is measured in ampere turns. The ampere turns are obtained by multiplying the number of turns of wire in coil, by the amperes of current flowing through it.

All forms of matter resist the development of magnetic force. This resistance is called magnetic reluctance. The reluctance of air is much greater than that of iron or steel, but is constant; that of iron and steel is not. If one thousand ampere turns develop a certain magnetic density in a circuit composed wholly of air, two thousand ampere turns will double this density. In iron and steel it will require much more than double the ampere turns to double the magnetic density.

If in a magnetic circuit ten inches long, 100 ampere turns develop a certain density, it will require 200 ampere turns to develop the same density if the magnetic circuit is double the length. The table on page 159 gives the ampere turns required to develop different magnetic densities in magnetic circuits one inch long, composed of air, iron and steel.

To find ampere turns required to develop any magnetic density in any magnet use following rule: —

Multiply the figures given in the table on page 159; for density required, by length of the magnetic circuit, and the product will be total number of ampere turns.

CHAPTER V.

SWITCH-BOARDS, DISTRIBUTING CIRCUITS AND SWITCH-BOARD INSTRUMENTS.

Generators of the constant potential type are made so as to develop a certain voltage at a given velocity, but in some cases it is not practicable to run them at the exact speed for which they are designed; and in others, it is desired to vary the voltage slightly, hence, all machines are provided with means for changing the e.m.f. slightly. This regulating device is also necessary in cases where the load is for a time light, and for the balance of the time heavy; for, as we have shown, the voltage will vary to some extent with changes in the strength of the current. If the generator is at some distance from the points where the current is used, the drop of voltage in the lines will be greater with strong currents; hence, when the load is heavy, it is necessary to increase the voltage developed by the generator. As it is not advisable to change the speed of the engine, the variation of voltage is obtained by changing the strength of the current that flows through the shunt field coils, and this is accomplished by providing a resistance that can be cut in or out of the shunt coil circuit, as is illustrated in Fig. 68, in which R represents the resistance, or field regulator, as it is called. When the lever is moved to the extreme left position, all the regulator resistance is cut out of the circuit, and then the voltage of the generator is the highest that can be obtained with the speed at which it is running. When the lever is moved to the extreme right, all the resistance of the regulator is introduced into the shunt coil circuit, and then the voltage is the lowest. By placing the lever in

intermediate positions between the extremes right and left, different voltages may be obtained.

To be able to operate a generator furnishing current to a system of distributing wires, it is necessary to have a number of

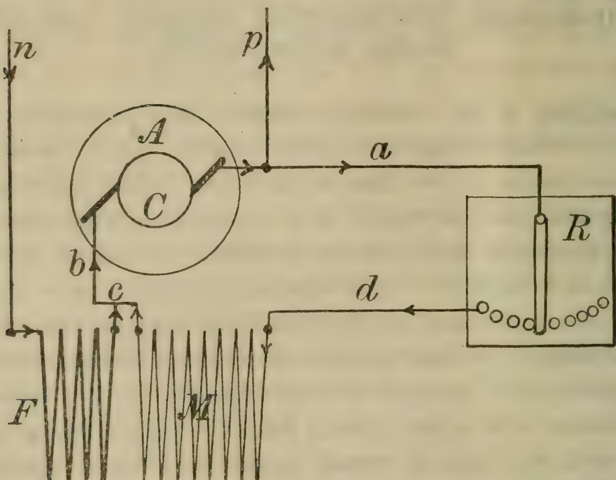


Fig. 68. Resistance regulator for shunt coil.

instruments and other devices, included in the circuit, some of which are absolutely indispensable, and others of which are simply conveniences, and may be looked upon as luxuries. The various devices required are shown in Fig. 69. The generator is shown at *M*, and at *e* the field regulator is placed, and it is connected with one of the generator armature terminals and with one end of the shunt coil wires by means of wires *d d*. The wires *c c* run from the generator terminals to the voltmeter *V*, and thus enable us to see what the voltage is at all times. Wires *a* and *b* convey the current to the external circuit, with which they can be connected or disconnected by means of switches *ss ss*. At *A* an ammeter is placed which indicates the strength of current in

amperes. The ammeter can be placed in either *a* or *b*, as the same strength current flows in both. At *f f* safety fuses are provided, so as to open the circuit in case the current becomes so strong as to be capable of overheating the generator wire. If one of the line wires runs out into the open air, and is carried along on poles, we will have to provide a lightning arrester, as shown at *h*, this being connected with the ground as at *g*. If both lines run into the open air, an arrester must be placed in both; and if both are confined to the interior of a building, no arresters will be required. From the points *m m* branch circuits may be run off in as many directions as necessary, and by providing switches *s s*, these can be connected or disconnected from the main line when desired.

This crude arrangement would enable us to operate the system successfully, but it would not be so convenient as a more methodi-

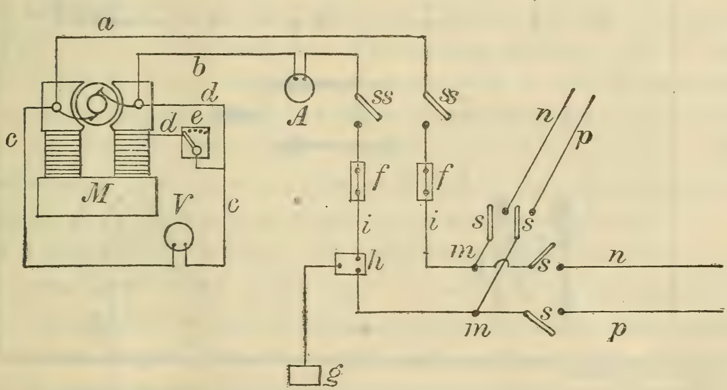


Fig. 69. Instruments required in the circuit.

cal grouping of the several devices and instruments. It represents the way things were done in the early days of electric lighting, but at the present time, instead of having the several parts scattered about in a helter-skelter fashion, they are all assembled

upon a large panel, which is called a switch-board. Fig. 70 gives the general arrangement of wiring and location of devices for a simple board arranged for one generator feeding into five external

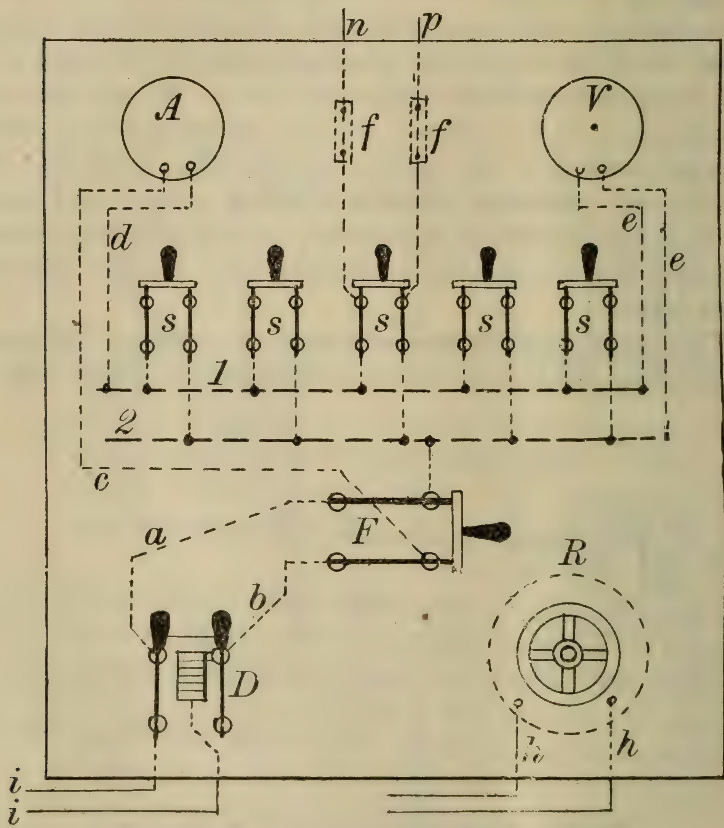


Fig. 70. General arrangement of switchboard.

circuits. The ammeter and voltmeter are placed at the top of the board, and directly under these are arranged five switches, *s*, which control the external circuits. One of these circuits is indi-

cated by the lines *n p*, *f f*, being safety fuses. The wires *i i* convey the main current from the generator to a circuit breaker *D*, which is simply a switch that is constructed so that it will open automatically when the current becomes too strong. From the circuit breaker, the current passes through wires *a* and *b* to the main switch *F*, and by wire *c*, it runs from here to the ammeter *A* and from the latter by wire *d* to a rod 1 which is called a bus bar. The upper side of the main switch is connected directly with bus 2. The voltmeter is connected with two busses by the wires *e e*. The field regulator is located back of the board at *R*, and is connected in the shunt coil circuit by means of wires *h h*. The switch of the regulator *R* is connected with a hand-wheel on the front of the switch-board, so that the attendant can watch the voltmeter as he turns the wheel and thus see just what affect the movement is producing on the voltage.

In addition to the devices shown in Fig. 70, we can, if desired, provide a recording ammeter, a recording voltmeter and a wattmeter; the first two would give us a record of the amperes and volts for a certain length of time, generally 24 hours, and the last one would register the amount of electrical energy. We could also provide ammeters for each one of the distributing circuits, so as to know the strength of current in each one.

If we desire to arrange the switch-board for two generators, and these are of the shunt type, we will require no changes in Fig. 70, except to provide another regulator and a main switch and circuit breaker for the additional machine. This arrangement of board is suitable for a single compound wound generator, or any number of shunt wound machines, but if we have two or more compound generators, the connections between these and the bus bars will have to be somewhat modified.

The modifications required in a switch-board for two or more compound generators can be made clear by the aid of Figs. 71 and 72. In the first figure, we can see that if the current return-

ing from the main line through n divides into wires a and b, it will remain divided until it passes through the armatures and the F coils of the two machines, and thence through wires e e, it will

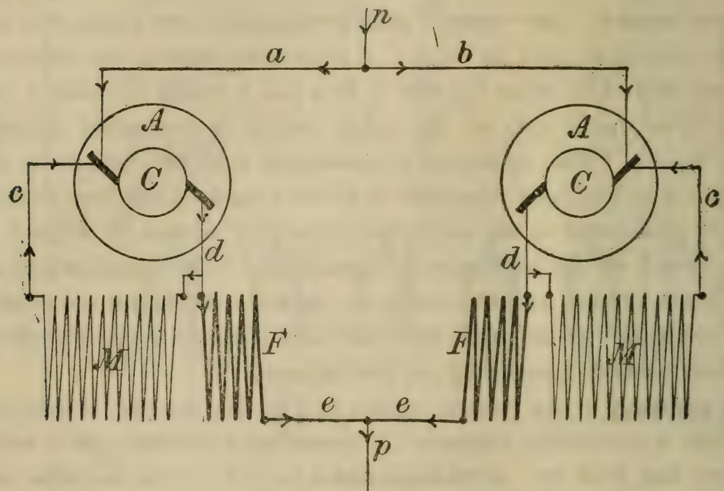


Fig. 71. Connections from machines to switchboard.

reunite again in wire p. In Fig. 72, the two parts of the current will flow through wires d d to the single wire e, and then divide into wires f f, and thus reach the coils F F, and, finally, through wires h h, reach p. In Fig. 71, if the right side armature generates more current than the other one, the F coil of that generator will be traversed by the strongest current, for in each machine the strength of current in the armature and the F coil will be nearly the same. Now, if the right side machine generates the strongest current, it is because its voltage is the highest, but the fact that its F coil will be traversed by the strongest current will make its voltage still higher, thus increasing the difficulty. In Fig. 72, the current flowing through the two F coils will be the same, no matter how much the two armature currents may differ,

for these come together in wire e, and passing from this to the two F coils, the current will divide in equal amounts. As can be seen, the effect of adding the wires d d, e and f f in Fig. 72 is to equalize the currents that flow through the F coils, and thus prevent, as far as possible, the unequal action of the generators.

When two or more compound generators are connected so as to feed into the same general circuit, the connections are made in accordance with Fig. 72. Fig. 73 illustrates a switchboard for two compound generators, and, as will be noticed, the most striking difference between it and Fig. 70, is that there are three bus bars instead of two. One of these busses is called the equalizer, and it takes the place of wires d d e and

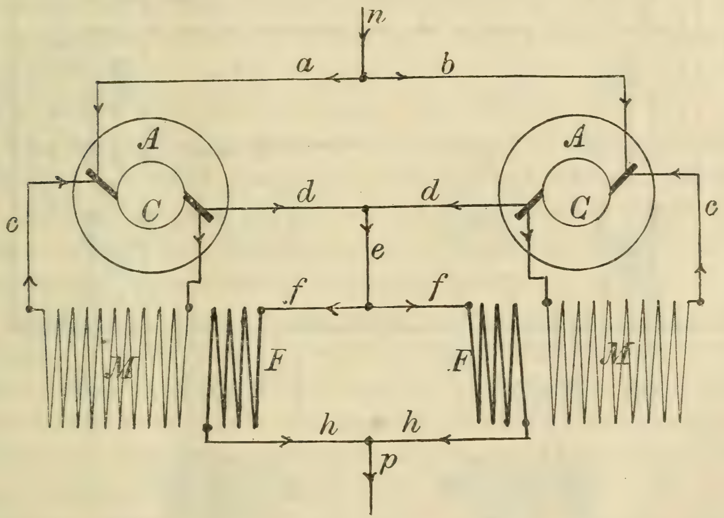


Fig. 72. Arrangement of equalizing connections.

f f in Fig. 72. The equalizing connections run from generator wires f to the main switches S, and thence to bus 1. The h wires of the generators run to one side of the circuit breakers D E,

and thence to the middle blades of the *S* switches, and from these to the bus 2. The generator wires run to the outside blades of

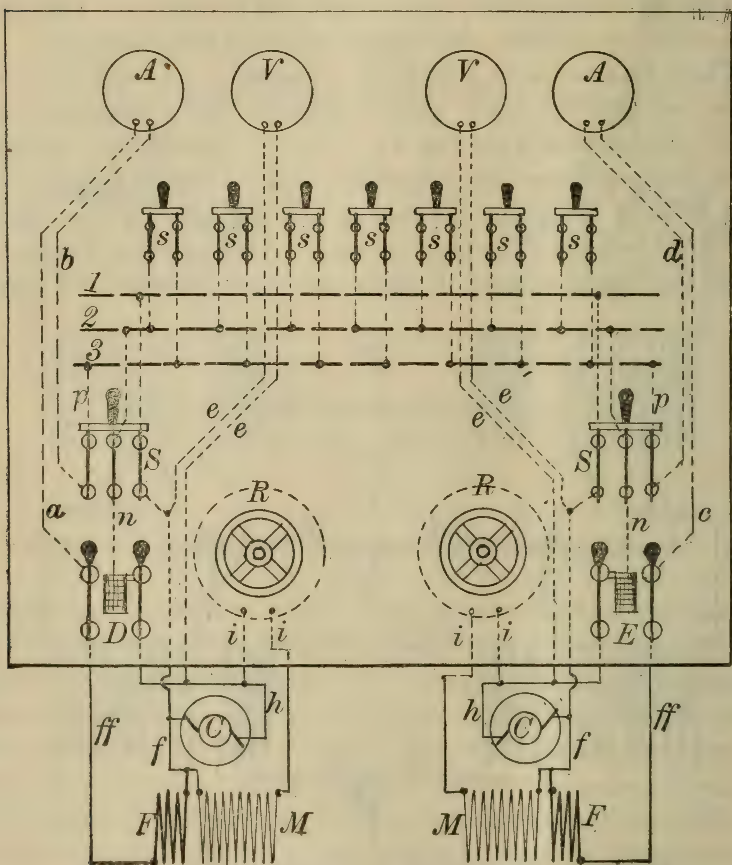


Fig. 73. Switchboard for two compound generators.

the circuit breakers, and from these to the ammeters *A A*, and thence to bus 3. The voltmeters are connected with wires *h* and *f*, and thus indicate the e.m.f.'s of the generators.

If another generator were added, it would be connected with the bus bars in the same way.

In starting two or more compound-wound generators, one machine is started first, and then the second is run up to full speed, and its voltage is adjusted by means of the regulator R, so as to be the same as that of the machine that is running. When the voltages of the two machines are equal, the main switch of the second machine is closed so as to connect it with the bus bars. This action will generally make a slight change in the voltage of the second machine, causing it to run up or down a trifle; and as a result by looking at the ammeters, we will find that it is taking more or less than its share of the load. If such is the case, we manipulate the regulator R, until the loads are properly divided. Whether the voltage of the second machine will rise or fall after it is connected with the bus bars, will depend upon the extent to which it is compounded; if slightly compounded, the voltage will drop, and if heavily compounded, it will rise.

The switch-boards illustrated are adapted to what is called the two-wire system of distribution, but in cases where it is desired to transmit the current to a considerable distance, without using extra large wire, the three-wire system of distribution is employed. This system is illustrated in Figs. 74 to 76.

Suppose we have two generators as indicated at G G in these diagrams, and let the direction of the current through both be from bottom toward the top; then it is evident, that if we remove the middle wire O, the lower machine will deliver current into the upper one, and if each generator develops an e.m.f. of 115 volts, the combined e.m.f. will be 230 volts, and this will be the pressure between the bottom and top wires; but the voltage between either wire and the center one will only be 115. Suppose we have a number of lamps connected between wire P and the center wire O, and an equal number of lamps between O and N, as is shown in Fig. 74; then it is evident that the same amount

of current will flow through both sets, and as a consequence, all the current that passes from the upper generator into wire *P* will go directly through both sets of lamps to the lower wire *N*, and thus return to the lower side of the bottom generator. Under

Fig. 74.

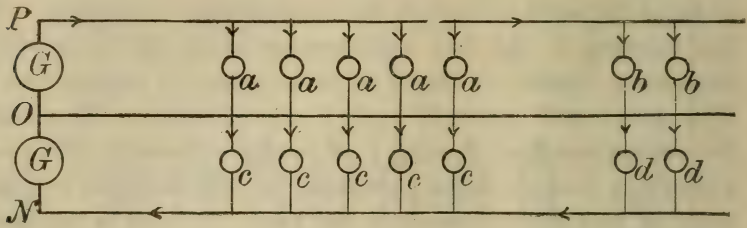


Fig. 75.

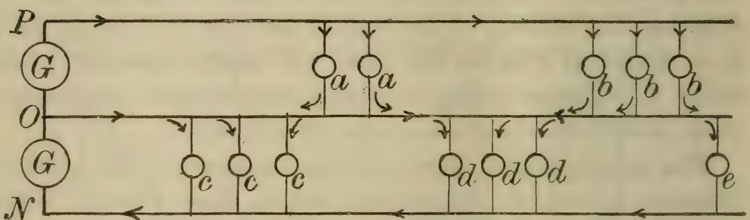
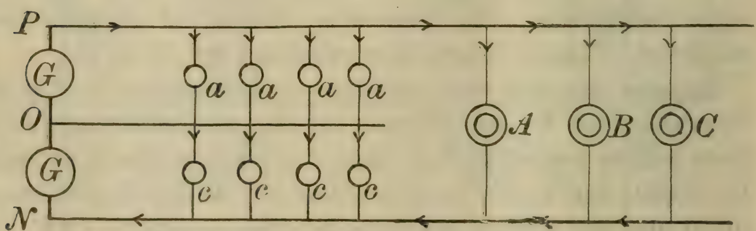


Fig. 76.



Arrangements of three-wire system.

these conditions, the lamps will be acted upon by 115 volts each, but the current will be driven through the circuit by a voltage of 230. Now, if the voltage is doubled, four times the number of lamps can be supplied with the same size wires; hence, the cost of line wire per lamp will be reduced to one-fourth. Suppose, that instead of having the lamps equally divided as in Fig. 74,

they are arranged as in Fig. 75; then since the current fed into the system from the upper wire P is only sufficient for five lamps, while there are seven lamps in the lower section, it follows that through wire O a current sufficient for two lamps must be supplied. The way in which the currents would flow in this case is clearly indicated by the arrows.

In Fig. 74, it will be seen that if we removed the middle wire, the lamps would not be affected, for none of the current comes through it; but in Fig. 75, if we cut the middle wire, two of the lower lamps would be unprovided for. From this it will be seen that the object of the middle wire is simply to provide the extra current required for the side that carries the largest number of lamps. If the lights are so arranged that on both sides of the central wire O the number is practically the same at all times, the center wire can be made very small, but such perfect balance cannot be obtained always, and on that accouut, the center, or neutral wire, as it is called, is made of the same size as the others, except in large systems, in which it is sometimes not more than one-third the size.

As motors require large amounts of current, they are nearly always made to operate with a voltage of 230, and are connected with the outside wires of the system, as is shown in Fig. 76, in which a a a a and c c c c indicate lamps connected between the sides and the neutral wire, and A B C are motors connected across the outside lines.

When a switch-board is arranged for two generators connected with a three-wire system, we use three bus bars, just as in Fig. 70, but discard the equalizing connection, and connect the generators with the busses in the same way as they are connected with wires N O and P in Figs. 74 to 76. If we have a number of generators feeding into the three-wire system, then we connect each set with an equalizer bus; that is, provide two sets of busses, and the P and N busses of these two sets we connect

with a third set in the proper order for the three-wire system, and from the latter busses the external circuits are fed.

If we desire to supply a larger building with a lighting and power system, we can run the wires in almost any way, providing we make connections with the lamps and motors, but if we adopt

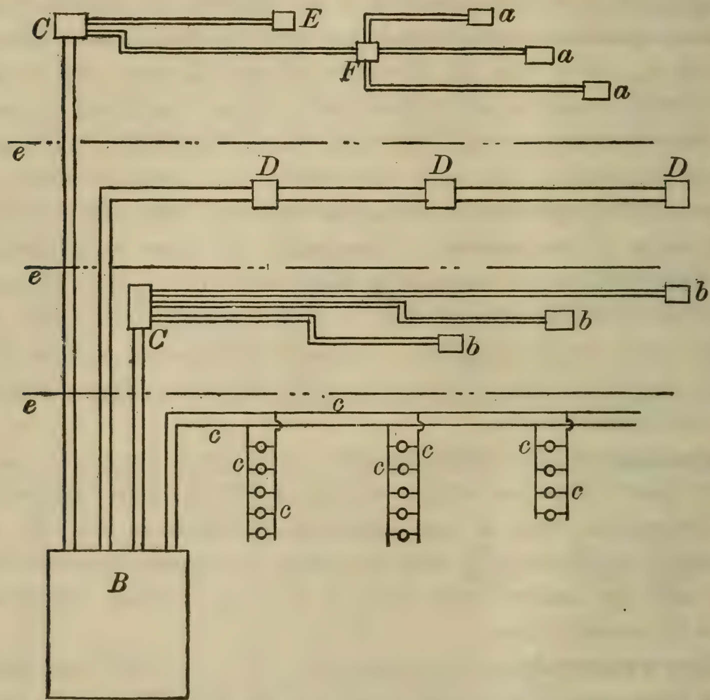


Fig. 77. Light and power system for building.

a systematic arrangement it will require less labor to operate the plant, and when anything goes wrong we will be able to locate the difficulty with much less trouble and in less time. The best way to accomplish this is by the use of small switch-boards located at different points in the building, these becoming centers

of distribution, from which all the lamps are supplied. The general arrangement of such a system can be understood from Fig. 77, in which *B* represents the main switch-board, located in the engine room, and *e e e* the several floors upon which the lights are located. From the main switch-board we run up four lines, one to each floor, and locate secondary boards at *C C* and *D D D*. We can also run out lines directly from the board to the lamp circuits as at *c c c c*. From the boards *C C*, we run circuits to smaller boards, as shown at *E*, *F*, *A*, *A*, *A*, and *b b b*. From each one of these small boards we can run out circuits to the lamps.

These small switch-boards are called panel boards or boxes, and also distribution boards. They are made of all sizes from eight or ten inches square, up to four or five feet, and are arranged to feed into one or two, or fifty or sixty circuits, supplying anywhere from five or six lights up to a thousand or more.

The construction of distribution boards can be understood from Figs. 78 and 79, the first being arranged for the three-wire system, and the second for the two-wire. Fig. 78 is arranged to feed ten circuits, and is provided with one main switch by means of which the entire box can be disconnected from the main line. The distributing circuits are provided with proper safety fuses on the outside wires, so that if anything goes wrong and the current increases to a dangerous point, the circuit will be open. No fuse is placed on the middle wire, as it is not necessary, and might result in cutting out both sides of the circuit when only one was disabled.

Fig. 79 is a more complete panel, because each one of the six distribution circuits is provided with a switch, so that it is possible to disconnect any of the circuits without interfering with the others. In some cases a distribution board of this kind is the only thing that will answer the purpose, but in others, the more

simple construction of Fig. 78 answers just as well. The fuses in Fig 78 are shown at *E F*. These fuses are sometimes made so that they can be used as switches; that is, they can be pulled out

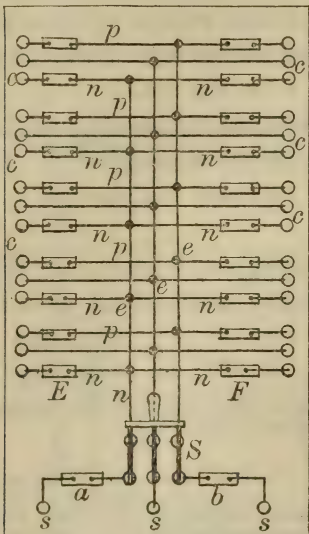


Fig. 78. Board for three-wire system.

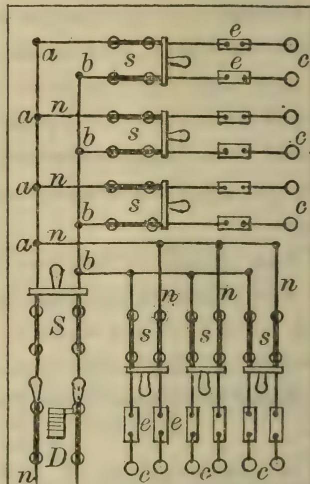


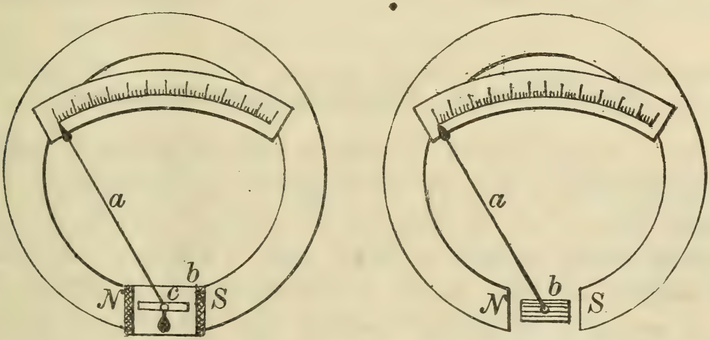
Fig. 79. Board for two-wire system.

of place and thus open the circuit, and if one blows out it can be removed and a new fuse be put in, and then it can be replaced, thus placing the disabled circuit in service without interfering with the others.

The ammeters and voltmeters used on switch-boards depend for their operation upon the repulsion between magnetic lines of force. A great many different constructions are used, but most of them operate upon the principles illustrated in Fig. 80 or 81. If a small bar of iron *c* is placed between the poles of a permanent magnet, as in Fig. 80, it will be held in the horizontal position by the attraction of the magnet. If it is surrounded by a stationary coil of wire *b*, through which a current of electricity passes, then

it will be under the influence of two forces, one the attraction of the poles $N S$ of the magnet, and the other the attraction of the lines of force developed by the current flowing through coil b. The action of the latter will tend to swing the rod c into the vertical position. The force of the magnet will remain constant, but the force of the coil will vary with the strength of the current passing through it; hence, the stronger the current the more the bar c will be swung around into the vertical position. If we provide a small counter-weight, as shown in the illustration, to resist the action of the coil, we will have a means that will enable us to adjust the movement of the bar, so that it will swing around through a given angle for a given increase in current. If a pointer a is secured to c it will swing over the scale as shown, when c is rotated by the action of the coil.

If coil b is mounted so that it may swing around the center pivot, we can discard bar c, for then as soon as a current traverses b, the lines of force developed around it will be attracted by

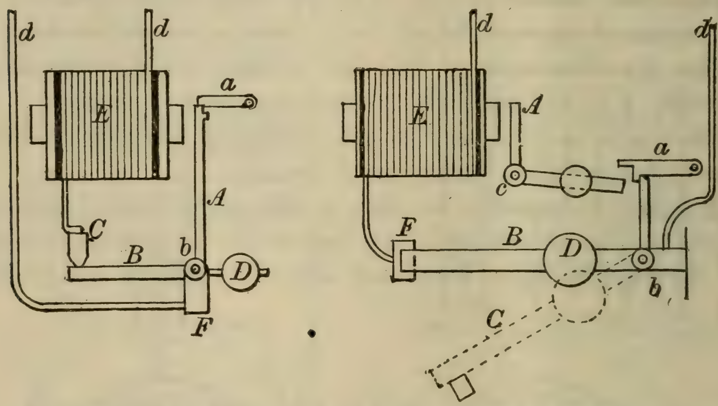


Figs. 80 and 81. Principles of ammeter and voltmeter.

those of the permanent magnet, and will exert a twisting force so as to place the axis of the coil parallel with the lines of force passing from N to S. In this case as in the previous case, the

effort to twist *b* around will be proportional to the strength of the current, hence, the stronger the current the greater the swing. Ammeters and voltmeters are made on these principles, and the only difference in the two instruments is in the size of the wire used for the coils.

Figs. 82 and 83 illustrate the principle upon which circuit breakers are made. In Fig. 82, suppose a current flows through magnet *E*, then it will attract the lever *A*, the latter being made



Figs. 82 and 83. Principles of circuit breakers.

of iron. If the current is weak it may not develop a sufficient attractive force in *E* to lift the weight *D*, and in that case *A* will remain where it is. If, however, the current is increased until *E* becomes strong enough to lift *D*, then *A* will move over toward the magnet, and the catch "*a*" falling behind it, will not allow it to return to its former position until placed there by hand. When *A* swings over, it carries *B*, and thus breaks the connection with *C* and opens the circuit. Thus it will be seen that by properly adjusting the weight *D* and the magnet *E*, we can set the device so as to open the circuit whenever the current reaches a certain strength. This is the principle upon which circuit break-

ers act, but such a device as Fig. 82 would be of no service for lighting circuits, because the distance by which *C* and *B* are separated is too small to break the current. By modifying the construction as in Fig. 83, we can obtain a device that will give a wide separation at the breaking point. In this construction, the lever *A* when drawn towards the magnet, strikes the catch *a*, so as to release lever *B*, and then the weight *D* throws the latter down to the position shown in broken lines, thus giving a wide separation between *F* and *C*. By moving the weight on the lower arm at *A*, the device can be adjusted so as to act with different strengths of current.

Circuit breakers as actually constructed, do not have the appearance of this diagram, but they operate on the principle illustrated by it.

The electromotive force in volts developed in the armature of a motor, or generator, can be determined if we know the number of wires upon the outer surface, the number of maxwells of magnetic flux that pass through the armature, and the revolutions per second. The rule for the calculation is as follows: —

Multiply the number of wires on the outer surface of the armature by the maxwells of magnetic flux and by the revolutions per second, and divide this product by 100,000,000.

This is the rule for two pole armatures. For multipolar armatures with series, or wave winding, use same rule making the flux equal to the sum of the fluxes issuing from all the positive poles.

For multipolar armatures with a lap, or parallel winding, use same rule but take the flux issuing from one pole only.

To obtain the pull in pounds of a motor armature at one foot radius use the following rule: —

Multiply the number of wires on the outer surface of armature by the amperes of armature current, and by total number of maxwells of magnetic flux passing through armature, and divide this product by 852,000,000. See pages 13 and 46.

CHAPTER VI.

ELECTRIC MOTORS.

Motors are made so as to run at a constant velocity, or for variable speed. For the latter type of machine, the field coils are wound in series, and for constant speed the shunt winding is used. A motor of either kind cannot be started successfully without placing an external resistance in the armature circuit, because, when the armature is at a standstill, there is nothing but the resistance of the wire to hold the current back, and as a result, if no extra resistance is provided, the first rush of current would be very great. As soon as the armature begins to revolve, an e.m.f. is induced in its wires, and this acts in opposition to the e.m.f. of the line current; that is, it acts like a back pressure, and holds the current back. On this account, the e.m.f. of a motor is called a counter e.m.f., and it is abbreviated into c.e.m.f.

The way in which the external resistance is connected with a motor is illustrated in Fig. 84, in which *M* is the motor and *R* the external resistance. *D* is a main switch, by means of which the motor is connected with the main line. This switch is closed first, and then switch *F* is moved to the right until it covers the first contact of the resistance *R*. The current can then pass directly to the field shunt coils through wire *e*, and thence by wire *a*, return to the main line. The armature current, however, has to first pass through the resistance *R*, before it can reach wire *i*, and thus the armature. As soon as the armature begins to speed up, the switch *F* is advanced, step by step, and in a few seconds it is moved to the extreme right position, in which all the resistance *R* is cut out of the armature circuit. When *F* reaches this position, the motor should be running at full speed.

If the current should stop while the motor is running, the machine would stop, also, and then, if the current were turned on again, the motor would be caught with the armature connected across the line without an external resistance, and as it would be at a standstill, the current would rise to an enormous strength. To prevent this, the switch *F* is always opened whenever the motor stops. The attendant may forget to do this, however; therefore automatic switches have been devised that will open themselves whenever the current dies out.

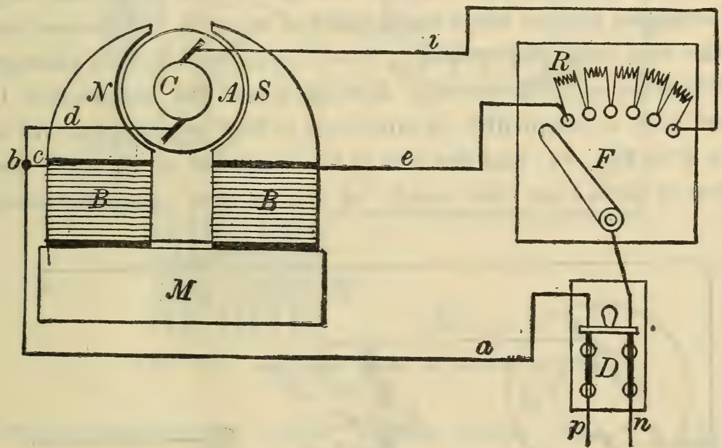


Fig. 84. External resistance connected with motor.

A simple switch provided with a resistance so as to be suited to start a motor, is called a motor-starter, and one that in addition is provided with means for automatically flying to the open position whenever the current fails, is called an automatic underload starter.

If the motor is very much overloaded, its speed will slow down and the current will increase in strength. If the overload is sufficient, the current will become so strong as to be able to burn out

the armature; hence, it is desirable to provide a circuit breaker that will open the circuit when the current becomes so strong as to be liable to burn out the machine. Motor-starters are made with a circuit-breaking attachment, and are then called automatic overload motor-starters. A device that combines the under and overload starter, features is called an automatic under and overload starter, and by some people it is called a "no voltage" and "overload starter."

When motors were first introduced, a great deal of trouble was experienced with the starters, owing to the fact that they were arranged so that when the motor was stopped, the connection with the field coils was broken. Now, the current flowing through the field coils objects to stop flowing when the connection is broken, and, consequently, it continues to flow between the end of switch F in Fig. 84, and the last of the contacts of R, until the distance is more than the e.m.f. of the current can overcome.

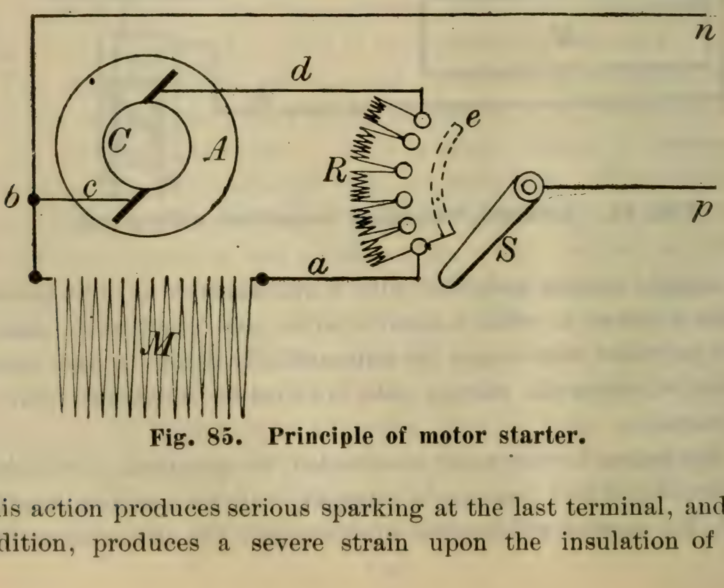


Fig. 85. Principle of motor starter.

This action produces serious sparking at the last terminal, and in addition, produces a severe strain upon the insulation of the

field coils, because, as the current is headed off in one direction, it tries to find an outlet in another. This action is what is

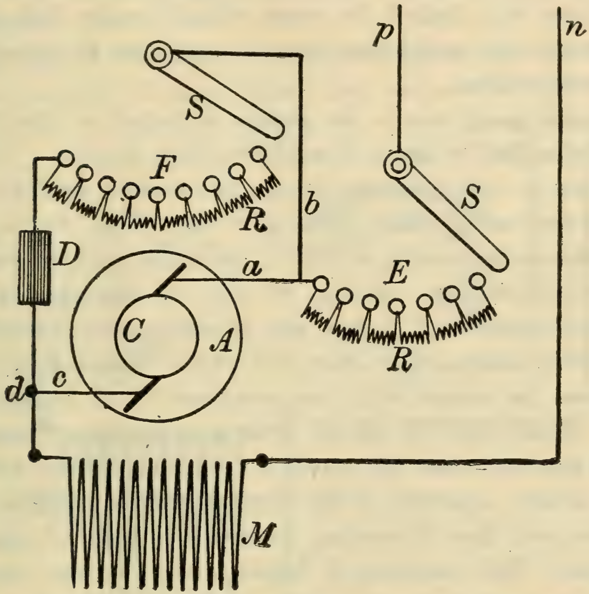


Fig. 86. Another style of motor starter.

commonly called the "kick of the motor field." All this trouble can be obviated by connecting the starter with the motor in such a way that the field circuit is never opened, as is shown in Fig. 84. As this is quite an important device it is presented in a more simple form in Fig. 85, in which it will be seen that the field coils and armature are permanently connected, so that when switch S opens the circuit, the field current can flow through the armature, until it dies out. All first-class concerns make motor starters with this connection, at the present time. Some of them add the curved contact e. Without this contact, it can be seen that when the switch S is moved to the top position, the

resistance R is simply transferred from the armature to the field circuit, and that the current going to the field coils has to pass through this resistance. As this resistance is insignificant in comparison with that of the shunt coils, it makes little difference whether it is left in the field circuit or not, but by the addition of e it can be cut out.

Variable speed motors are always arranged so that the speed may be changed by hand as conditions may require. Trolley-car motors are of this type, and so are the motors used for printing presses, and many other kinds of work. Figs. 86 to 88 show arrangements by means of which the speed may be varied with series wound motors. In Fig. 86, E is the starting box and F is the speed regulator. In the act of starting, the switches are in the position shown. To start, the switch S and E is turned so as to close the circuit with the resistance R all included. S is moved toward the left as the armature speeds up, and reaches the last position when full speed is attained. If the switch of F is now closed, a portion of the current will be diverted from the armature, and thus its rotating force will be reduced, and thereby its speed. This method of speed control is also arranged so that the two switches act together, so as to introduce resistance into the motor circuit, and at the same time divert more or less of the current around the armature. It is not used extensively, as all the current that passes through F is just so much thrown away.

In Fig. 87 the speed is controlled by means of the switch F, which cuts out portions of the field coils and this changes the strength of the field. With this arrangement, if a portion of the field is cut out, the motor will run faster, because the c.e.m.f will be reduced, therefore, the armature current will be increased. To obtain a wide range of regulation, it is necessary to wind a large number of turns of wire on the field, so that with all the wire in service, the speed may be the lowest required.

Fig. 88 shows another arrangement that varies the strength of the field by diverting a portion of the current through switch *F*. It gives as wide a range of regulation as Fig. 87, but is not so economical.

Figs. 86 and 88 cannot be used to regulate the speed of shunt motors, but Fig. 87 can. The first two named figures, if used

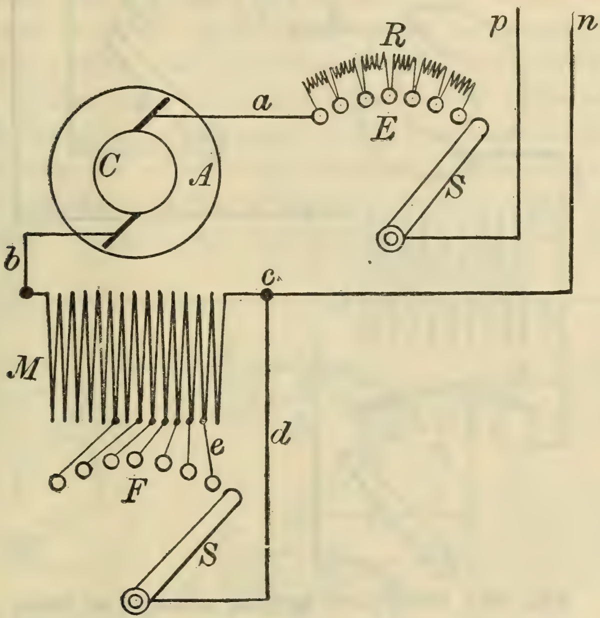


Fig. 87. Regulator for shunt motor.

with a shunt motor, would simply afford a third path through which current could pass from one side of line to the other, that is, from the *p* to the *n* wires, but this would not materially affect the strength of current that would flow through the armature and field coils. They work with series wound motors, because the current is not shunted from wire *p* to wire *n* but simply from one side of the armature, or the field, to the other.

Fig. 89 shows an arrangement by means of which a shunt motor can be made for variable speed. In this case, the switch

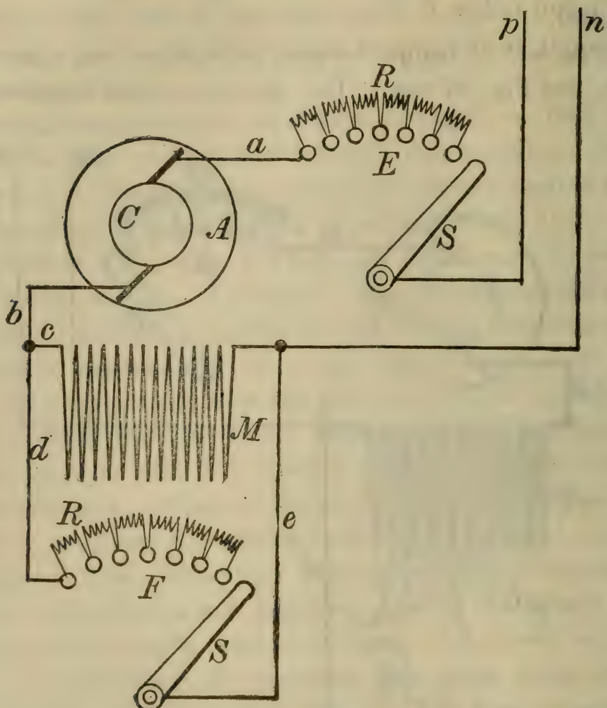


Fig. 88. Device for varying strength of field.

and resistance *E* is simply an ordinary starter, and *F* is a resistance that is introduced in the field circuit, so as to vary the strength of the field. With this arrangement the slowest speed is obtained when all the resistance of *F* is out of the circuit.

The direction in which a motor runs can be reversed by simply reversing the direction of the current through the armature. Any of the arrangements for varying the speed can be used in connection with reversible motors by arranging the switch so as

to reverse the armature connections. Fig. 90 will give a fair idea of the way in which a reversing switch is made. This represents the type of switch used most generally for this purpose, and it is known as the cylinder switch. It is the kind used on trolley-cars. The vertical row of circles numbered from one to eleven represents stationary contact pieces, to which the terminals of the motor, the line and the resistance are attached. The shaded parts *B B* are metal plates that are secured to the surface of a cylinder, that is so located that as it is turned in one direction or the other, these plates slide over the stationary contacts. If the cylinder is turned so that the plates on the right side slide over the contacts, the motor will run in one direction, and if the cylinder is turned in the other direction, the motor will be reversed. Suppose the right side plates slide over the con-

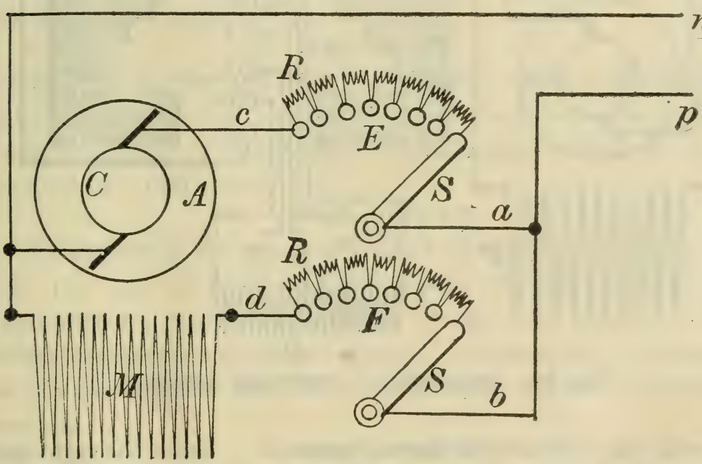


Fig. 89. Device for varying speed of shunt motor.

tacts, then the current from p will pass to contact 2, and thence to wire a, and to the left-side of the field. Through wire d it will return from the field to contact 5, and by means of

plates N and T, which are connected as shown at X^1, it will reach contact 3 and wire b, which runs to the lower side of the armature. From the top of the armature, through wire c, the current will return to contact 4 and through plates S and M and the connection X will reach contact 6, which by one of the wires e con-

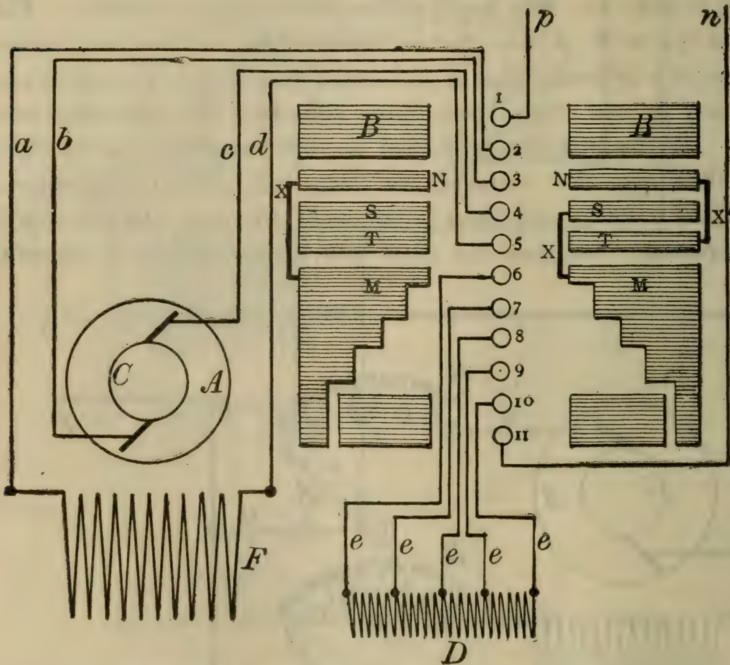


Fig. 90. Principle of reversing device.

nects with the left-side of the resistance D. From the right-side of this resistance, the current will pass to contact 10, and thus to contact 11, through the cylinder plate, and in that way reach line wire n.

If the cylinder is turned further around, contact 7 will be covered by plate M, and this will cut one section of D. By a further

movement, contact 8 will be covered, thus cutting out another section, and by continuing the movement, all of D can be cut out.

If the cylinder is turned so as to slide the left-side plates over the contacts, the change effected will be that contact 5 will be connected with 4 instead of with 3, and contact 6 will be connected with 3 instead of 4, thus reversing the direction of the current through the armature.

The strength of an electric current is measured in amperes. The electromotive force that drives an electric current through a circuit is measured in volts. The resistance that a wire or other circuit offers to the passage of an electric current through it is measured in ohms.

The unit of resistance, the ohm, is the resistance of a column of mercury about 40 inches long and about five hundredths of an inch in diameter, or, to be more exact, 106 centimeters long, and one millimeter in diameter.

THE WATT.

The watt is the unit of electric power — the volt-ampere, the power developed, and is equal to $\frac{1}{746}$ of one horse power. A convenient multiple of this is called the Kilowatt, written K. W., and is equal to 1,000 watts.

THE AMPERE.

The ampere is the practical unit of electric current, such a current [or rate of flow, or transmission of electricity] as would pass, with an electromotive force of one volt, through a circuit whose resistance is equal to one ohm; a current of such a strength as would deposit from solution .006084 grains of copper per second.

CANDLE POWER.

The candle power is the unit of light; and a standard candle is a candle of definite composition which with a given consumption in a given time, will produce a light of a fixed and definite brightness. A candle which burns 120 grains of spermaceti wax per hour, or two grains per minute, will give an illumination equal to one standard candle.

CHAPTER VII.

INSTRUCTIONS FOR INSTALLING AND OPERATING SLOW AND MODERATE SPEED GENERATORS AND MOTORS.

To remove the armature, take off the brush-holders, brush yoke, pulley and bearing caps and put a sling on the armature, as shown in accompanying illustration. A spreader of suitable length should be used and its location adjusted to prevent the rope from drawing against the flange or end connections.

In assembling, marked parts of the machine should be assembled strictly according to the marking. Clean all connection joints carefully before clamping them together. Wipe the shaft-bearing sleeves and oil cellars perfectly clean and free from grit. Place the bearing sleeves and oil rings in position on the shaft and then lower the armature into place, taking care that the oil rings are not jammed or sprung. As soon as the armature is in position, pour a little oil in the bearing sleeves, put the caps on the boxes and screw them down snugly. The top field should next be put on and bolted firmly into position, and a level placed on the shaft to check the leveling of the foundation.

Fill the bearings with the best grade of thin lubricating oil and do not allow it to overflow. Oil throwing is usually due to an excess of oil and can be avoided by care in filling the oil cellars.

To complete the assembly, place the pulley on the shaft, draw up the set screws and put on the brush rigging and connection blocks.

STARTING.

Before putting on the belt, see that all screws and nuts are tight and turn the armature by hand to see that it is free and

does not rub or bind at any point. Put on the belt with the machine so placed on the rails as to have the minimum distance between pulley centers. Start the machine up slowly and see that the oil rings in bearings are in motion. As the machine

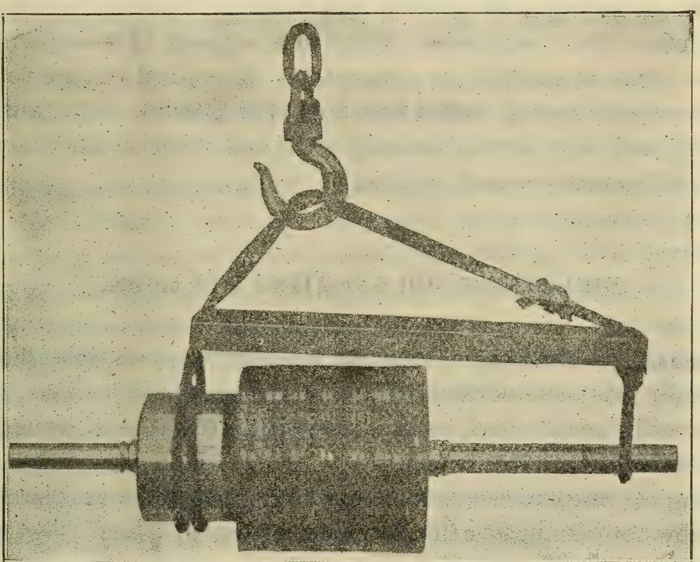


Fig. 91. Method of raising an armature.

comes up to speed, tighten the belt till it runs smoothly, and run the machine long enough without load to make sure that the bearings are in perfect condition. The bearings, when running, should be examined at least once a week.

CARE OF COMMUTATOR.

The commutator brushes and brush-holders should at all times be kept perfectly clean and free from carbon or other dust. Wipe the commutator from time to time with a piece of canvas lightly coated with vaseline. Lubricant of any kind should be applied very sparingly.

If a commutator when set up begins to give trouble by roughness, with attendant sparking and excessive heating, it is necessary to immediately take measures to smooth the surface. Any delay will aggravate the trouble, and eventually cause high temperatures, throwing off solder, and possibly displacement of the segments. No. 0 sandpaper, fitted to a segment of wood, with a radius equal to that of the commutator, if applied in time to the surface when running at full speed (and if possible with brushes raised), and kept moving laterally back and forth on the commutator, will usually remedy the fault.

DIRECTIONS FOR STARTING DYNAMOS.

General. — Make sure that the machine is clean throughout, especially the commutator, brushes, electrical connections, etc. Remove any metal dust, as it is very likely to make a ground or short circuit.

Examine the entire machine carefully, and see that there are no screws or other parts that are loose or out of place. See that the oil-cups have a sufficient supply of oil, and that the passages for the oil are clean and the feed is at the proper rate. In the case of self-oiling bearings, see that the rings or other means for carrying the oil work freely. See that the belt is in place and has the proper tension. If it is the first time the machine is started, it should be turned a few times by hand, or very slowly, in order to see if the shaft revolves easily and the belt runs in center of pulleys.

The brushes should now be carefully examined and adjusted to make good contact with the commutator and at the proper point, the switches connecting the machine to the circuit being left open. The machine should then be started with care and brought up to full speed, gradually if possible; and in any case

the person who starts either a dynamo or a motor should closely watch the machine and everything connected with it, and be ready to throw it out of circuit if it is connected, and shut down and stop it instantly if the least thing seems to be wrong, and should then be sure to find out and correct the trouble before starting again.

STARTING A DYNAMO.

In the case of a dynamo it is usually brought up to speed either by starting up a steam-engine or by connecting the dynamo to a source of power already in motion. The former should, of course, only be attempted by a person competent to manage steam-engines and familiar with the particular type in question. This requires special knowledge acquired by experience, as there are many points to consider and attend to, the neglect of any of which might cause serious trouble. For example, the presence of water in the cylinder might knock out the cylinder-head; the failure to set the feed of the oil-cups properly might cause the piston-rod, shaft, or other part, to cut. And other great or small damage might be done by ignorance or carelessness. The mere mechanical connecting of a dynamo to a source of power is usually not very difficult; nevertheless, it should be done carefully and intelligently, even if it only requires throwing in a friction-clutch or shifting a belt from a loose pulley. To put a belt on a pulley in motion is difficult and dangerous, particularly if the belt is large or the speed is high, and should not be tried except by a person who knows just how to do it. Even if a stick is used for this purpose, it is apt to be caught and thrown around by the machinery, unless it is used in exactly the right way.

It has been customary to bring dynamos to full speed before the brushes are lowered into contact with the commutator; but

this is not necessary, provided the dynamo is not allowed to turn backwards, which sometimes occurs from carelessness in starting, and might injure copper brushes by causing them to catch in the commutator. If the brushes are put in contact before starting, they can be more easily and perfectly adjusted and the e.m.f. will come up slowly, so that any fault or difficulty will develop gradually and can be corrected; or the machine can be stopped, before any injury is done to it or to the system. In fact, if the machine is working alone on a system, and is absolutely free from any danger of short-circuiting any other machine or storage battery on the same circuit, it may be started while connected to the circuit, but not otherwise. If there are a large number of lamps connected in the circuit, the field magnetism and voltage might not be able to "build up" until the line is disconnected an instant.

If one dynamo is to be connected with another, or to a circuit having other dynamos or a storage battery working upon it, the greatest care should be taken. This coupling together of dynamos can be done perfectly, however, if the correct method is followed, but is likely to cause serious trouble if any mistake is made.

SWITCHING DYNAMOS INTO CIRCUIT.

Two or more machines are often connected to a common circuit. This is especially the case in large buildings where the number of lamps required to be fed varies so much that one dynamo may be sufficient for certain hours, but two, three or more machines may be required at other times. The various ways in which this is done depending upon the character of the machines and of the circuit and the precautions necessary in each case make this a most important and interesting subject, which requires careful consideration.

Dynamos may be connected together either in parallel (multiple arc) or in series.

DYNAMOS IN PARALLEL.

In this case the + terminals are connected together or to the same line, and the — terminals are connected together or to the other line. The currents (i. e. amperes) of the machines are thereby added, but the e.m.f. (volts) are not increased. The chief condition for the running of dynamos in parallel is that their voltages shall be equal, but their current capacities may be different. For example: A dynamo producing 10 amperes may be connected to another generating 100 amperes, provided the voltages agree. Parallel working is, therefore, suited to constant potential circuits. A dynamo to be connected in parallel with others or with a storage battery, must first be brought up to its proper speed, e.m.f., and other working conditions, otherwise, it will short-circuit the system, and probably burn out its armature. Its field magnetism must, therefore, be at full strength, owing to the fact that it generates no e.m.f. with no field magnetism. Hence, it is well to find whether the pole pieces are strongly magnetized by testing them with a piece of iron, and to make sure of the proper working of the machine in all other respects before connecting the armature to the circuit. It is a common accident for the field-circuit to be open at some point, and thus cause very serious results. In fact, a dynamo should not be connected to a circuit in parallel with others until its voltage has been tested and found to be equal to, or slightly (not over 1 or 2 per cent) greater than that of the circuit. If the voltage of the dynamo is less than that of the circuit, the current will flow back into the dynamo and cause it to be run as a motor. The direction of rotation is the same, however, if it is shunt-wound, and no great harm results from a slight difference of potential. But a compound-wound machine requires more careful handling.

DIRECTIONS FOR RUNNING DYNAMOS AND MOTORS.

In the case of a machine which has not been run before, or has been changed in any way, it is of course wise to watch it closely at first. It is also well to give the bearings of a new machine plenty of oil at first, but not enough to run on the armature, commutator or any part that would be injured by it, and to run the belt rather slack until the bearings and belt have gotten into easy working condition. If possible a new machine should be run without load or with a light one for an hour or two, or several hours in the case of a large machine; and it is always wrong to start a new machine with its full load, or even a large fraction of it.

This is true even if the machine has been fully tested by its manufacturer and is in perfect condition, because there may be some fault in setting it up, or some other circumstance which would cause trouble. All machinery requires some adjustment and care for a certain time to get it into smooth working order.

When this condition is reached, the only attention required is to supply oil when needed, keep the machine clean and see that it is not overloaded. A dynamo requires that its voltage or current should be observed and regulated if it varies. The person in charge should always be ready and sure to detect the beginning of any trouble, such as sparking, the heating of any part of the machine, noise, abnormally high or low speed, etc.; before any injury is caused, and to overcome it by following directions given elsewhere. Those directions should be pretty thoroughly committed to mind, in order to facilitate the prompt detection and remedy of any trouble when it suddenly occurs, as is apt to be the case. If possible, the machine should be shut

down instantly when any trouble or indication of one appears, in order to avoid injury and give time for examination.

Keep all tools or pieces of iron or steel away from the machine while running, as they might be drawn in by the magnetism, and perhaps get between the armature and pole-pieces and ruin the machine. For this reason, use a zinc, brass or copper oil-can instead of iron or "tin" (tinned iron).

Particular attention and care should be given to the commutator and brushes to see that the former keeps perfectly smooth and that the latter are in proper adjustment. (See Sparking).

Never lift a brush while the machine is delivering current, unless there are one or more other brushes on the same side to carry the current, as the spark might make a bad burnt spot on the commutator.

Touch the bearings and field-coils occasionally to see that they are not hot. To determine whether the armature is running hot, place the hand in the current of air thrown out from it by centrifugal force.

Special care should be observed by any one who runs a dynamo or motor to avoid *overloading* it, because this is the cause of most of the troubles which occur.

BATTERIES.

Name of Cell.	E.M.F.	Material in Plates.		Electrolyte.
Bunsen . .	1.95	Zinc.	Carbon.	Nitric and Sulphuric Acid.
Grove . . .	1.93	"	"	" " " "
Gravity . .	1.06	"	Copper.	Copper Sulphate. Zinc "
Leclanche .	1.45	"	Carbon.	Sal Ammoniac.
Dry Cell . .	1.49	"	"	Sal Ammoniac Paste.
EdisonLelande	0.9	"	CopperOxide	Caustic Potash.
Lead Storage .	1.98	Lead.	Lead.	Sulphuric Acid.

CHAPTER VIII.

WHY COMMUTATOR BRUSHES SPARK AND WHY THEY DO NOT SPARK.

A long list of reasons might be given why commutator brushes spark, and why they do not spark, but by such a procedure no light would be thrown on the subject, because the reasons would not be understood unless fully explained. It is preferable to explain the subject and let the reader tabulate the reasons after digesting the explanation of the principles involved.

Whenever an electric current is interrupted, a spark is produced and it makes no difference how the current is generated, or what kind of a conductor it is flowing through. To break a current without a spark is not a possibility; hence, if we desire to open a circuit without producing a spark, the only way to accomplish the result is by killing the current before the circuit is opened. The brushes resting on the commutator of a motor or a generator have to transmit to the armature and take away from it the current that is generated, in the case of a generator, or the current that drives the machine in the case of a motor. If the brushes were made so narrow that they could only make contact with one commutator segment at a time, it would be impossible to run the machine without producing very destructive sparks. Commutators, however, are not made in this way. The insulation between the segments is narrow, and the brushes are wide enough to be always in contact with two segments, and part of the time with three. Suppose that the proportions are such that during most of the time the brush only touches two

segments, as shown in Fig. 92. With these proportions it will be seen, that so long as there are two segments in contact with each brush, it is a possibility for the current to be diverted through one of them only. Suppose that at the instant when the forward segment is passing from under the brush, all the current flows through the rear segment; then it is quite evident that the first-named segment will break away from contact with the brush without making a spark, for there will be no current flowing from it to the brush.

All the foregoing is self-evident, but it will be suggested that although the brush can break away from the front segment without producing a spark, it cannot do the same thing with the rear segment, for all the current is flowing through this one. While it is true that when the forward segment passed from under the brush all the current was flowing through the rear segment, it is not true that the current continues to follow this path. As soon as the front segment passes from under the brush, the rear one becomes the forward segment, and while it is advancing to the point where it must pass from under the brush, the current can be transferred to the next segment back of it which now plays the part of rear segment. Thus we see that to be able to run a machine without producing sparks at the commutator, all we have to do is to provide means whereby the current is transferred from one segment to the one back of it as the commutator revolves, so that when the segments pass from under the brush there is no current flowing through them. This result is accomplished more or less perfectly in all machines, made by responsible firms. There are machines on the market that have been designed by men who are not well enough posted in the principles of electrical science to obtain proper proportions, and these are not proportioned so as to shift the current from the forward to the rear segment as fast as the machine revolves; such machines always produce more or less serious sparking.

If a machine is accurately made and the armature coils and commutator segments are properly spaced and sufficient in number, it is possible to get the brushes so there will be little or no spark at a given load; but if the current strength be increased or reduced, the sparks will appear, and the more the current is changed the larger the sparks will be, the increasing current producing the greatest sparking.

The way in which the current is shifted from the front to the rear segment will be explained in connection with Fig. 92. In this figure, A represents a portion of the core of a ring armature. The shaft upon which it is mounted is shown at D, and P N are the corners of the poles between which it rotates. The small blocks C represent a portion of the commutator segments, which we have placed outside of the armature, so as to make the diagram as simple as possible. For the same reason we have shown the armature coils as made of two turns of wire each. The line F divides the space between the ends of the poles into two equal parts, and the line E divides the armature into two halves on which the directions of the induced currents is opposite. In all the coils to the right of line E the currents are induced in a direction away from the shaft, and in all the coils to the left of line E the currents flow toward the shaft, all of which is clearly indicated by the arrow heads placed upon the lines representing the coils. The outline B represents the end of one of the brushes, and from the direction in which it is inclined it will be understood that the armature revolves in a direction counter to that of the hands of a clock.

When the armature is in the position shown, the current flowing in the coils to the right of line E passes to segment b, and thus reaches the brush, while the current flowing in the coils to the left of line E reaches segment a, and through this passes to the brush. As the brush rests upon segments a and b the coil with which they connect is short-circuited, and therefore a

current can flow in it in any direction, or there may be no current. To be able to run without spark, or to obtain perfect commutation, as it is called, the current in this short-circuited coil, when in the position shown, should be zero, or nearly so. This coil, which is short-circuited by the brush, is called the commutated coil, or the coil undergoing commutation. It will be noticed that this commutated coil is in a position just forward of

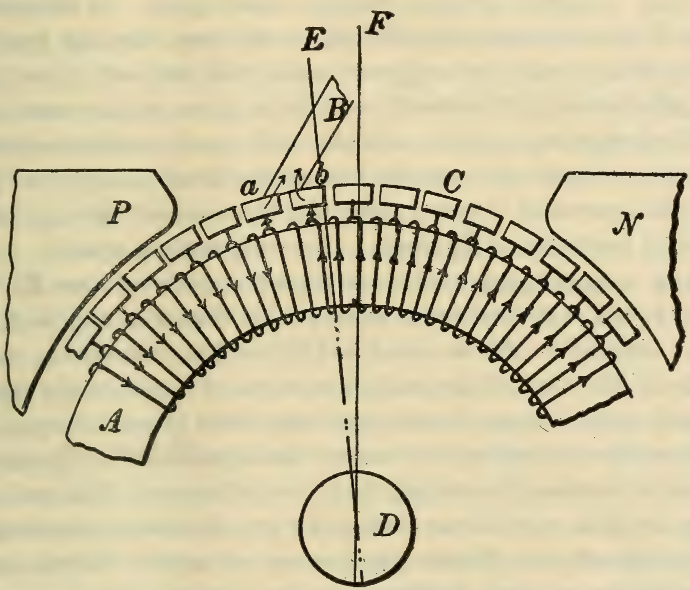


Fig. 92. Portion of core of ring armature.

the line *E*; hence, the action of pole *P* will be to develop a current in it that will flow in the same direction as the current in the coils ahead of it, that is, in the coils to the left. Now if this current flowed through the brush, it would be in a direction contrary to that of the arrow at *a*; hence it would act to check the current flowing from the front segment *a* to the brush, and would divert it through the commutated coil to the rear segment

b. If the action of pole *P* upon the commutated coil is sufficiently vigorous, the current developed in it will be as strong as the current in the coils ahead of it, by the time the end of the segment is about to break away from the brush; and this being the case there will be no current from segment *a* to the brush, and consequently, no spark. If the action of pole *P* is not strong enough, then there will be a small current from segment *a* to the brush when they separate, and as a result, a small spark. If the action of pole *P* on the commutated coil is too vigorous, then the current developed in it will be too great, and it will not only divert all the current coming from the forward coils, through the commutated coil to segment *b*, but in addition will develop a local current that will circulate through the end of the brush, and, therefore, when the separation occurs, there will be a current flowing from the brush to the front segment, and consequently a spark.

If the commutated coil were placed astride of line *E*, the action of pole *P* upon it would be no greater than that of pole *N*, so that no current would be developed in it while undergoing commutation. The farther the coil is in advance of line *E*, when short-circuited by the brush, the stronger the action of pole *P* upon it; therefore, the strength of the current developed in the commutated coil can be increased by moving the brush farther away from pole *P*. Hence, by trial, a point can be found where the current developed will be just sufficient for the purpose and no more. This is true, supposing the current developed by the armature to remain constant, but, if it varies, the current in the commutated coil will be either too great or too small. If, when the brushes are set, the armature is delivering a current of, say, twenty amperes, then the current flowing through the coils to the left of the brush will be ten amperes, and the current in the commutated coil will also be ten amperes. If the armature current increases to forty amperes, the current in the forward coils will be twenty amperes, and as that in the commutated coils will still be ten amperes, it will have only

one-half the strength required for perfect commutation. In practice, however, it is found that if the commutator have a sufficient number of segments, and the proportions of the machine are such that the line E remains practically in the same position for all strengths of armature current, then, if the brushes are set so as to run sparkless with an average load, they will run so nearly sparkless with a heavy or light load as to make it difficult to detect the difference.

Even when a machine is properly proportioned, the brushes may spark badly if they are not set in the proper position and with the proper tension. If the tension is not right, they will spark no matter where they are set. If the tension is too light, they will spark, because they will chatter and thus jump off the surface of the commutator. If the tension is too great, they will spark because they will cut the commutator, and then the latter will act as a file or grindstone and cut away particles from the brushes, and these will conduct the current from segment to segment, as well as from the segment to the brush. Whenever a commutator is seen to be covered with fine sparks, some of which run all the way around the circle, it may be depended upon that the surface is rough, due in most cases to too much pressure on the brushes, and the remedy is to smooth it up and reduce the tension and set the brushes where they will run with the smallest spark. When the brushes begin to spark they rarely cure themselves and the longer they are allowed to run with a heavy spark the worse they will get.

Of all the troubles which may occur, sparking is the only one which is very different in the different types of machines. In some its occurrence is practically impossible. In others, it may result from a number of causes. The following cases of sparking apply to nearly all machines, and they cover closed-coil dynamos and motors completely.

Cause 1. — Brushes not set at the neutral point.

Symptom. — Sparking, varied by shifting the brushes with rocker-arm.

Remedy. — Carefully shift brushes backwards or forwards until sparking is reduced to a minimum.

The usual position for brushes in two-pole machines is opposite the spaces between the pole-pieces.

Cause 2. — Commutator rough, eccentric, or has one or more "high bars" projecting beyond the others, or one or more flat bars, commonly called "flats," or projecting mica, any one of which causes brush to vibrate, or to be actually thrown out of contact with commutator.

Symptom. — Note whether there is a glaze or polish on the commutator, which shows smooth working; touch revolving commutator with tip of finger-nail, and the least roughness is perceptible, or feel of brushes to see if there is any jar. If the machine runs at high-voltage (over 250), the commutator or brushes should be touched with a small stick or quill to avoid danger of shock. In the case of an eccentric commutator, careful examination shows a rise and fall of the brush when commutator turns slowly, or a chattering of brush when running fast.

Remedy. — Smooth the commutator with a fine file or fine sandpaper, which should be applied by a block of wood which exactly fits the commutator (in latter case, be careful to remove any sand remaining afterward; and *never use emery*). If bearing is loose put in new one. If commutator is very rough or eccentric, it should be taken out and turned off.

Cause 3. — Brushes make poor contact with commutator.

Symptom. — Close examination shows that brushes touch only at one corner, or only in front or behind, or there is dirt on surface of contact. Sometimes, owing to the presence of too much oil or from other cause, the brushes and commutator become very dirty and covered with smut. They should then be carefully cleaned by wiping with oily rag or benzine, or by similar means.

Occasionally a "glass-hard" carbon brush is met with. It is incapable of wearing to a good seat or contact and will only touch in one or two points, and should be discarded.

Remedy. — File, bend, adjust or clean brushes until they rest evenly on commutator, with considerable surface of contact and with sure but light pressure.

It sometimes happens that the brushes make poor contact, because the brush-holders do not turn or work freely.

Cause 4. — Short-circuited coil in armature or reversed coil.

Symptom. — A motor will draw excessive current, even when running free without load. A dynamo will require considerable power even without any load.

The short-circuited coil is heated much more than the others, and is very apt to be burnt out entirely; therefore, stop machine immediately. If necessary to run machine to locate the short-circuit, one or two minutes is long enough, but it may be repeated until the short-circuited coil is found by feeling the armature all over.

An iron screw-driver or other tool held between the field-magnets near the revolving armature vibrates very perceptibly as the short-circuited coil passes. Almost any armature, particularly one with teeth, will cause a slight but rapid vibration of a piece of iron held near it, but a short-circuit produces a much stronger effect only *once* per revolution.

The current pulsates and torque is unequal at different parts of a revolution, these being particularly noticeable when armature turns rather slowly. If a large portion of the armature is short-circuited, the heating is distributed and harder to locate. In this case a motor runs very slowly, giving little power, but having full-field magnetism.

Remedy. — A short circuit is often caused by a piece of solder or other metal getting between the commutator bars or their connections with the armature, and sometimes the insulation between

or at the ends of these bars is bridged over by a particle of metal. In any such case the trouble is easily found and corrected. If, however, the short-circuit is in the coil itself, the only real cure is to rewind the coil.

One or more " grounds " in the armature may produce effects similar to those arising from a short circuit.

Cause 5. — Broken circuit in armature.

Symptom. — Commutator flashes violently while running, and commutator-bar nearest the break is badly cut and burnt; but in this case no particular armature coil will be heated, as in the last case and the flashing will be very much worse, even when turning slowly. This trouble, which might also be confounded with a bad case of " high-bar " or eccentricity in commutator (sparking), is distinguished from it by slowly turning the armature, when violent flashing will continue if circuit is broken, but not with eccentric commutator or even with " high bar."

Remedy. — The trouble is often found where the armature wires connect with the commutator and not in the coil itself, and the break may be repaired or the loose wire may be resoldered or screwed back in place. If the trouble is due to a broken commutator connection and it cannot be fixed, then connect the disconnected bar to the next by solder, or " stagger " the brushes; that is, put one a little forward and the other back so as to bridge over the break. If the break is in the coil itself, rewinding is generally the only cure.

Cause 6. — Weak field-magnetism.

Symptom. — Any considerable vibration is almost sure to produce sparking, of which it is a common cause. This sparking may be reduced by increasing the pressure of the brushes on the commutator, but the vibration itself should be overcome by the remedies referred to above.

Cause 7. — Chatter of Brushes. The commutator sometimes

becomes sticky when carbon brushes are used, causing friction, which throws the brushes into rapid vibration as the commutator revolves, similarly to the action of a violin-bow.

Symptom. — Slight tingling or jarring is felt in brushes.

Remedy. —Clean commutator and oil slightly. This stops it at once.

NOISE.

Cause 8. — Vibration due to armature or pulley being out of balance.

Symptom. — Strong vibration felt when the hand is placed upon the machine while it is running. Vibration changes greatly if speed is changed.

Remedy. — The easiest method of finding in which direction the armature is out of balance is to take it out and rest the shaft on two parallel and horizontal A-shaped metallic tracks sufficiently far apart to allow the armature to go between them. If the armature is then slowly rolled back and forth, the heavy side will tend to turn downward. The armature and pulley should always be balanced separately. An excess of weight on one side of the pulley and an equal excess of weight on the opposite side of the armature will not produce a balance while running, though it does when standing still; on the contrary, it will give the shaft a strong tendency to "wobble." A perfect balance is only obtained when the weights are directly opposite, i. e., in the same line perpendicular to the shaft.

Cause 9. — Armature strikes or rubs against pole pieces.

Symptom. — Easily detected by placing the ear near the pole-pieces, or by examining armature to see if its surface is abraded at any point, or by examining each part of the space between armature and field, as armature is slowly revolved, to see if any

portion of it touches or is so close as to be likely to touch when the machine is running. Or turn armature by hand when no current is on, and note if it sticks at any point.

Remedy. — Bind down any wire, or other part of the armature that may project abnormally, or file out the pole-pieces where the armature strikes, or center the armature so that there is a uniform clearance between it and the pole-pieces at all points.

Cause 10. — Singing or hissing of brushes. This is usually occasioned by rough or sticky commutator, or by tips of brushes not being smooth, or the layers of a copper brush not being held together and in place. With carbon brushes, hissing will be caused by the use of carbon which is gritty or too hard. Vertical carbon brushes, or brushes inclined against the direction of rotation, are apt to squeak or sing. A new machine will sometimes make noise from rough commutator, no matter how carefully it is turned off, because the difference in hardness between mica and copper causes the cut of the tool to vary, thus forming inequalities which are very minute, but enough to make noise. This can be best smoothed by running.

Remedy. — Apply a *very little* oil or vaseline to the commutator with the finger or a rag. Adjust the brushes or smooth the commutator. Carbon brushes are apt to squeak in starting up, or at slow speed. This decreases at full speed, and can usually be reduced by moistening the brush with oil, care being taken not to have any drops, or excess of oil. Shortening or lengthening the brushes sometimes stops the noise. Run the machine on open circuit until commutator and brushes are worn smooth.

For alternating current machinery and principles of alternating current see page 815.

HEATING IN DYNAMO OR MOTOR.

General Instructions. — The degree of heat that is injurious or objectionable in any part of a dynamo or motor is easily determined by feeling the various parts. If the heat is bearable for a few moments, it is entirely harmless. But if the heat is unbearable for more than a few seconds, the safe limit of temperature has been passed, except in the case of commutators in which solder is not used ; and it should be reduced in some of the ways that are given above. In testing with the hand, allowance should always be made for the fact that bare metal feels much hotter than cotton, etc. If the heat has become so great as to produce an odor or smoke, the safe limit has been far exceeded and the current should be shut off and the machine stopped immediately, as this indicates a serious trouble, such as a short-circuited coil or a tight bearing. The machine should not again be started until the cause of the trouble has been found and positively overcome. Of course neither water nor ice should ever be used to cool electrical machinery, except possibly the bearings of large machines, where it can be applied without danger of wetting the other parts.

Feeling for heat will answer in ordinary cases, but of course, the sensitiveness of the hand differs, and it makes a very great difference whether the surface is a good or bad conductor of heat. The back of the hand is more sensitive and less variable than the palm for this test. But for accurate results a thermometer should be applied and covered with waste or cloth to keep in the heat. In proper working the temperature of no parts of the machine should rise more than 45° C., or 81° F. *above* the temperature of the surrounding air. If the actual temperature of

the machine is near the boiling point, 100° C., or 212° F., it is seriously high.

It is very important in all cases of heating to locate correctly the source of heat in the exact part in which it is produced. It is a common mistake to suppose that any part of a machine which is found to be hot is the seat of the trouble. A hot bearing may cause the armature or commutator to heat or vice versa. In every case, all parts of the machine should be felt to find which is the hottest, since heat generated in one part is rapidly diffused throughout the entire machine. It is generally much surer and easier in the end to make observations for heating by starting with the whole machine perfectly cool, which is done by letting it stand for one or more hours or over night, before making the examination. When ready to try it, run it fast for three to five minutes, with the field magnets charged; then stop, and feel all parts immediately. The heat will be found in the right place, as it will not have had time to diffuse from the heated to the cool parts of the machine. Whereas, after the machine has run some time, any heating effect will spread until all parts are equal in temperature, and it will then be almost impossible to locate the trouble.

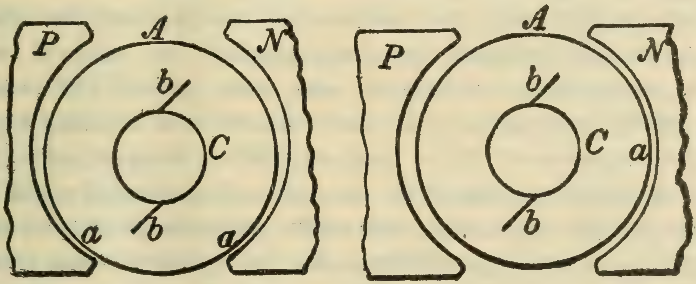
Excessive heating of commutator, armature, field magnets, or bearings may occur in *any* type of dynamo or motor, but it can almost always be avoided by proper care and working conditions.

THE EFFECT OF THE DISPLACEMENT OF THE ARMATURE.

If a machine is old, it is more than likely the shaft will be found out of center, and if this fact is discovered at a time when things are not working as they should, it is taken for granted this is the cause of the trouble. What is true of shafts out of the

center is true of several other things that are liable to get out of place. For the present it will be sufficient to investigate just what effect the displacement of the shaft can have.

Fig. 93 illustrates an armature of a two-pole machine which is out of center in one direction, and Fig. 94 shows another two-pole armature out of center in a direction at right angles to that shown in the first figure. The condition shown in Fig. 93 could be produced by a heavy armature running in rather light bearings for several years, and the side displacement of Fig. 94 could be produced by the tension of an extra tight belt. The mechan-



Figs. 93 and 94. Showing armature out of center.

ical effect of both these conditions would be to increase the pressure on the bearings, as the part *a* of the armature would be drawn toward the poles of the field with greater force than the opposite side. The downward pull, due to the attraction of the magnetism, would be greater in Fig. 93 than the side pull in Fig. 94 supposing both armatures and fields to be the same in both cases, and the displacement of the shafts equal. This difference is due to the fact that in Fig. 93 the magnetism of both poles is concentrated at the lower corners on account of the shorter air gap; hence both sides pull much harder on the lower side. In

Fig. 94 the pull of the *N* pole is greater than that of the other, simply because in the latter the magnetism is more dispersed, but the difference in the density on the two sides will not be very great. If the bearings of a machine, with the armature displaced, as indicated, have shown any signs of cutting, or if they run unusually warm, their condition will be improved by putting in new bearings that will bring the shaft central.

If the armature is of the drum type, the displacement of the shaft will have no effect upon it electrically. This is owing to the fact that all the armature coils are wound from one side of the core to the other, and, therefore, at all times, every coil has one side under the influence of one pole and the other side under the influence of the opposite pole, and if one side is acted upon strongly by one pole, it will be acted upon feebly by the other. If the armature is of the ring type, then the displacement of the shaft will affect it electrically, for in a ring armature, the coils on one side are acted upon by the pole on that side, only, and as the magnetic field from one pole will be stronger than that from the other (that is, considering the action upon equal halves of the armature), the voltage developed in the coils on one side of the armature will be greater than that developed on the other side.

The effect of the disturbance of the electrical balance will be that the brushes will spark badly, because the voltage of the current generated on one side of the armature will be greater than that of the current on the other side. Hence, when these two currents meet at the brushes, the strong one will tend to drive the weak one backward. If, while the armature is out of center, we wish to adjust the brushes so as to get rid of the excessive sparking, all we have to do is to set them to the right of the center line, as in Fig. 94 so that the wire on the left side will cover a greater portion of the circumference than the right.

In a multipolar machine, the displacement of the armature will have the same effect mechanically as in the two-pole type; multipolar armatures are connected in two different ways, one of which is called the wave or series winding, and the other the lap or parallel winding. In the first named type of winding, the ends of all the coils on the armature are connected with each other and with the commutator segments in such a manner that there are only two paths through the wire for the current; therefore, these two armature currents pass under all the poles and the voltage of each current is the combined effect of all the poles. From this very fact, it can be clearly seen that it makes no difference what the distance between the several poles and armature may be, for if some are nearer than the others, the only effect will be that these poles will not develop their share of the total voltage, but whatever their action may be, it will be the same on the coils in both circuits.

When a multipolar armature is connected so as to form a parallel or lap winding, then the connections between the coil ends, and between these ends and the commutator segments, are such that as many paths are provided for the current as there are poles, and each one of these paths is located under one pole, and as a consequence, the voltage developed in it is proportional to the action of this pole. The diagram Fig. 95 illustrates a six-pole armature with the ends of the field poles, and the arrows *a a*, *b b*, *c c*, indicate the six separate divisions of the coils in which the branch currents are developed. Now, it can be clearly seen that as the armature is nearer to the lower poles than to any of the others, the action of these will be the strongest. Hence, the currents *a a* will be stronger than the others and will have a higher voltage.

The two upper currents are weaker than the side ones and

their voltage is also lower, so that, the current returning to the commutator through the brushes at the upper corners, will not divide equally, but the larger portion will be drawn into the coils on the side; and as the upper coils will have to fight to hold their own, so to speak, there will be a disturbance of the balance that

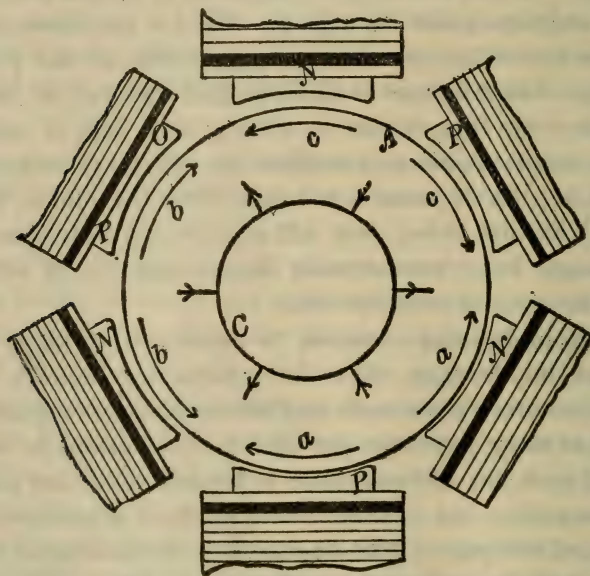


Fig. 95. Diagram of six-pole armature.

is required for smooth running. The result will be heavy sparking at these brushes. In the great majority of cases, if the brushes of a multipolar machine spark on account of the armature being out of center, the only cure is to reset the bearings, if they are adjustable, and if they are not, to put in new ones.

Table of Carrying Capacity of Wires. — Below is a table which must be followed in placing interior conductors, showing the allowable carrying capacity of wires and cables of ninety-eight per cent conductivity, according to the standard adopted by the American Institute of Electrical Engineers.

B. & S. G.	TABLE A. Rubber-Covered Wires. Amperes.	TABLE B. Weatherproof Wires. Amperes.	Circular Mils.
18	3	5	1,624
16	6	8	2,583
14	12	16	4,107
12	17	23	6,530
10	24	32	10,380
8	33	46	16,510
6	46	65	26,250
5	54	77	33,100
4	65	92	41,740
3	76	110	52,630
2	90	131	66,370
1	107	156	83,690
0	127	185	105,500
00	150	220	133,100
000	177	262	167,800
0000	210	312	211,600
Circular Mils.			
200,000	200	300	
300,000	270	400	
400,000	330	500	
500,000	390	590	
600,000	450	680	
700,000	500	760	
800,000	550	840	
900,000	600	920	

Circular Mils.	TABLE A. Rubber-Covered Wires. Amperes.	TABLE B. Weatherproof Wires. Amperes.
1,000,000	650	1,000
1,100,000	690	1,080
1,200,000	730	1,150
1,300,000	770	1,220
1,400,000	810	1,290
1,500,000	850	1,360
1,600,000	890	1,430
1,700,000	940	1,490
1,800,000	970	1,550
1,900,000	1,010	1,610
2,000,000	1,050	1,670

The lower limit is specified for rubber-covered wires to prevent gradual deterioration of the high insulations by the heat of the wires, but not from fear of igniting the insulation. The question of drop is not taken into consideration in the above tables.

Insulation Resistance. — The wiring in any public building must test free from grounds; i. e., the complete installation must have an insulation between conductors and between all conductors and the ground (not including attachments, sockets, receptacles, etc.) of not less than the following: —

Up to	5 amperes,	4,000,000	Up to	200 amperes,	100,000
"	10 "	2,000,000	"	400 "	25,000
"	25 "	800,000	"	800 "	25,000
"	50 "	400,000	"	1,600 "	12,500
"	100 "	200,000			

All cutouts and safety devices in place in the above.

Where lamp sockets, receptacles and electroliers, etc., are connected, one-half of the above will be required.

Soldering Fluid. — *a.* The following formula for soldering fluid is suggested: —

Saturated solution of zinc chloride,	5 parts.
Alcohol,	4 parts.
Glycerine,	1 part.

Bell or Other Wires. — *a.* Shall never be run in same duct with lighting or power wires.

Table of Capacity of Wires. —

B. & S. G.	Area Actual C. M.	No. of Strands.	Size of Strands B. & S. G.	Amperes.
19	1,288	...	...	...
18	1,624	...	...	3
17	2,048	...	...	...
16	2,583	...	...	6
15	3,257	...	...	...
14	4,107	...	...	12
12	6,530	...	...	17
...	9,016	7	19	21
...	11,368	7	18	25
...	14,336	7	17	30
...	18,081	7	16	35
...	22,799	7	15	40
...	30,856	19	18	50
...	38,912	19	17	60
...	49,077	19	16	70
...	60,088	37	18	85
...	75,776	37	17	100
...	99,064	61	18	120
...	124,928	61	17	145
...	157,563	61	16	170

B. & S. G.	Area Actual C. M.	No. of Strands	Size of Strands B. & S. G.	Amperes.
...	198,677	61	15	200
...	250,527	61	14	235
...	296,387	91	15	270
...	373,737	91	14	320
...	413,639	127	15	340

When greater conducting area than that of B. & S. G. is required, the conductor shall be stranded in a series of 7, 19, 37, 61, 91 or 127 wires, as may be required; the strand consisting of one central wire, the remainder laid around it concentrically, each layer to be twisted in the opposite direction from the preceding.

TABLE SHOWING THE SIZE OF WIRE OF DIFFERENT METALS THAT WILL BE MELTED BY CURRENTS OF VARIOUS STRENGTHS.

Strength of Current in Amperes.	DIAMETER OF WIRE IN THOUSANDTHS OF AN INCH.					
	Copper.	Aluminum.	Platinum.	German Silver.	Iron.	Tin.
1	.002	.003	.003	.003	.005	.007
2	.003	.004	.005	.005	.008	.011
3	.004	.005	.007	.007	.010	.015
4	.005	.006	.008	.008	.012	.018
5	.006	.008	.010	.010	.014	.021
10	.009	.012	.016	.016	.022	.033
15	.013	.016	.020	.020	.028	.044
20	.015	.019	.025	.025	.034	.053
25	.018	.022	.029	.029	.040	.062
30	.020	.025	.032	.032	.045	.069
35	.022	.028	.036	.036	.050	.077
40	.025	.030	.039	.039	.055	.084
50	.027	.033	.042	.042	.059	.091
60	.029	.035	.045	.045	.063	.098

CHAPTER IX.

INSTRUCTIONS FOR INSTALLING AND OPERATING APPARATUS FOR ARC LIGHTING, BRUSH SYSTEM.

Theory of the Brush arc generator. — The Brush Arc Generator is of the open coil type, the fundamental principle of which is illustrated in Fig. 96. Two pairs of coils, placed at right angles

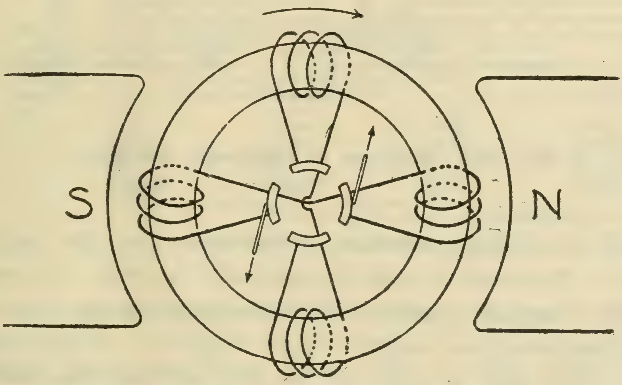


Fig. 96. Diagram of Brush arc dynamo.

on an iron core, are rotated in a magnetic field. The horizontal coils represented in the diagram are producing their maximum electromotive force, while the pair of coils at right angles to them is generating practically no electromotive force. The brushes are placed on the segments of the four-part commutator, so as to collect only the current generated by the two horizontal coils. The other coils are open circuited, or completely cut out of the circuit.

Such a machine will generate current, continuous in direction, but fluctuating considerably in amount. These fluctuations will be diminished by the addition of more coils to the armature.

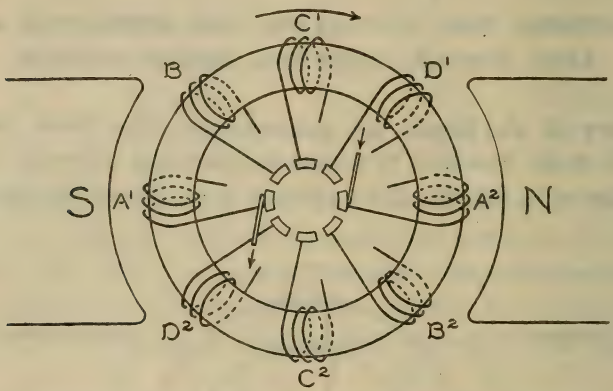


Fig. 97. Diagram of eight-coil machine.

Fig. 97 is a diagrammatic representation of an eight-coil machine. The ends of coils diametrically opposite are connected as in the four-pole machine, but to avoid complications, these connections have been omitted on the diagram. In the eight-coil machine, one pair of coils, A^1, A^2, is generating maximum electromotive force. At right angles to these coils, the coils C^1 and C^2 are generating no electromotive force. In intermediate positions, the coils B^1, B^2, D^1, D^2 are generating a useful electromotive force, although one which is not so high as that generated by the coils A^1 and A^2.

In collecting the current from such an armature, the coils in the intermediate positions cannot be connected in parallel with the coils generating maximum electromotive force, because their electromotive force is lower. The pair of coils A^1, A^2 can, however, be placed in series and connected in series with the two pairs

of coils B^1, B^2 and D^1, D^2, which may be placed in parallel with each other, since they occupy similar positions in the magnetic field.

In the Brush Arc Generator a double commutator is used to automatically make these connections.

In Fig. 98 this commutator is developed or spread out, and the coils are represented diagrammatically.

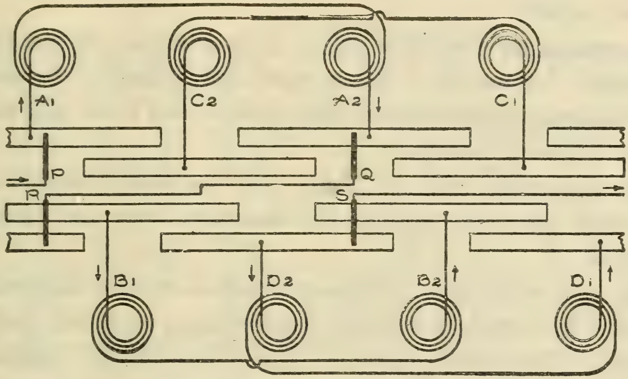


Fig. 98. Showing commutator spread out.

BIPOLAR BRUSH ARC GENERATORS.

Bipolar brush machines were built in eight sizes, ranging in capacity from 1 to 65 lamps of 2000 candle-power, and 2 to 45 lamps of 1200 candle-power.

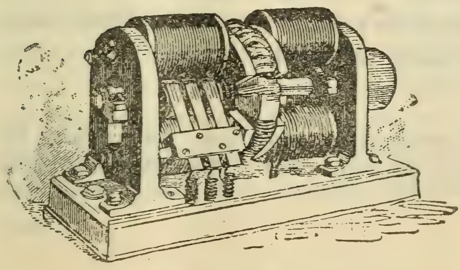


Fig. 99. Brush dynamo.

Although now superseded by the larger multipolar machines, so many bipolar machines are still in use that we consider it advisable to publish the following in-

structions for operating and making such repairs as become necessary after the long service which thousands of these machines have undergone.

The general construction of the bipolar machine is shown on page 105. Four field spools are provided, one pair to each side of the armature. The field cores are bolted to vertical yokes at each end of the machine, which also carry the bearings for the armature shaft.

The machines should be set up in the manner described under Multipolar Generators. To operate satisfactorily, the machines must be kept perfectly clean, the oil cups well filled and the commutator surfaces smooth.

The armature with its shaft may be readily removed after unscrewing the bolts and lifting the caps from the bearings at each end.

Each coil or bobbin on the armature is wound independently, and may be rewound without disturbing any other part of the armature. The inside ends of coils diametrically opposite are connected together, while their opposite ends are connected by means of wires running through the hollow shaft to opposite segments of the commutator.*

Proper connections are made by having separate brushes for each commutator ring consisting of two pairs of segments. Thus in an eight-coil machine, the lower brush on one commutator ring is connected to the upper brush on the next ring.

The commutator segments are mounted on an insulated body, and when worn out may be easily replaced.

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* That is for 2000 and 1200 candle-power machines. On machines for 4000 candle-power, each pair of opposite coils is connected in multipie instead of as above described.

CONNECTIONS OF NO 7½ AND NO. 8 BIPOLAR GENERATORS.

The field switch on the No. 7½ and No. 8 machines is different from that of the smaller sizes, and there is but one small binding post for connection to the regulator. The internal connections of the regulator are also slightly different. The inside

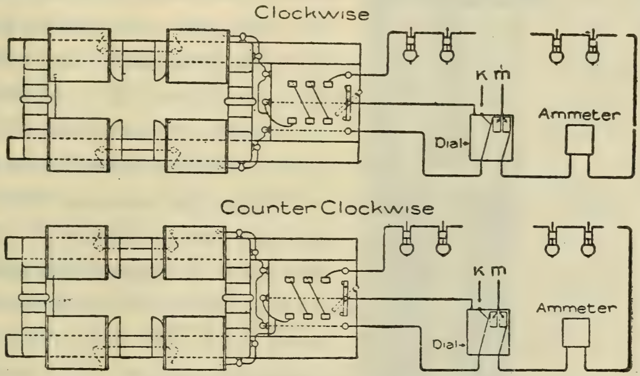


Fig. 100. Showing connections of bipolar machines.

terminal of one upper binding post *K* (see Fig. 100), is connected to the positive (left-hand) wire, which connects the main binding post to the magnets *M*.

As the No. 7½ and No. 8 machines have three commutator rings, cross-connections between the brushes are required as shown in the diagram. The outside left-hand brush is connected across to the middle right-hand brush, and the middle left-hand brush is connected to the inside right-hand brush. The inside left-hand brush is connected to the fields, and to the shunt leading to the regulator. On the old type machines, the inside left-hand brush is connected to the right-hand small binding post.

AUTOMATIC REGULATOR FOR BIPOLAR BRUSH ARC GENERATORS.

The Brush automatic regulator or "Dial" is shown in the accompanying illustration. It contains a variable resistance which is connected as a shunt to the fields, and automatically changed to increase or decrease the field current, and thus the voltage of the machine. The resistance is composed of columns of carbon plates which rest on the lever L.

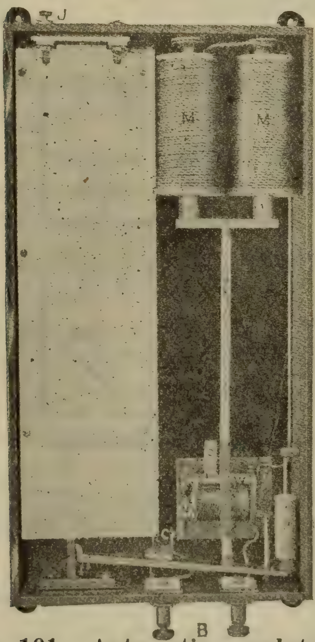


Fig. 101. Automatic regulator.

When the current rises above normal, the magnets M draw up the lever L and compress the carbon columns, reducing their resistance and shunting more current from the fields. The electromotive force of the generator is thus reduced and the current maintained constant. As the resistance of the line is increased by the addition of lamps, the current in the magnets M is diminished and the lever drops, separating the carbons and increasing the resistance of the shunt. More of the current must then pass through the generator fields and raise the electromotive force of the machine.

The dash pot P is to prevent sudden changes of resistance, and should be kept full of pure cylinder oil or glycerine. It should move easily, so that the regulator can respond quickly to changes in the current.

The variable resistance W, which is adjusted by the spring S, is connected as a shunt to the magnet coils M and regulates their current. The opening of the contact C is adjusted by the fiber nut N. With the shunt resistance properly adjusted and the lever in a midway position, the current can be increased by tightening the nut N, and decreased by loosening it. When the shunt resistance W is once properly adjusted, it should not be changed, unless the magnets M are changed.

To connect the regulator into the circuit, the main line is brought in at the large binding posts B, the positive being connected to the left-hand binding post. The current *must enter* at the left hand. See Fig. 101.

The small binding post J is connected with the small binding post on the generator, so that the carbon resistance plates are in shunt with the field of the generator.

While adjusting the regulator, the generator should run at normal speed, and the first test should be made on short circuit. If the current is too low, lever L should begin to drop at once and so increase the current. If this lever should stand at its lowest position and the current still remain too low at full load, the speed of the generator must be increased. If the current is too high, the lever should rise and so reduce it; if it fails to rise, the contact C should be examined. The current at this contact should spark all the time. If the lever rises when the contact is below normal the nut N should be tightened.

The resistance of the shunt W should be so adjusted by removing the spring S that the lever will rise when the contact is opened and descend when the contact is closed.

Once in six months the carbon resistance columns should be loosened up and the dust and loose carbon particles blown out with a bellows. The regulator must then be readjusted.

MULTIPOLAR BRUSH ARC GENERATORS.

Each machine is provided with an iron bed-plate fitted with a ratchet and screw for sliding the machine to adjust the belt tension. This bed-plate should be securely fastened to a dry wood sub-base not less than 10″ in thickness, except on wood floors, in which case it may be somewhat less, according to the thickness of the floor. See Fig. 102.

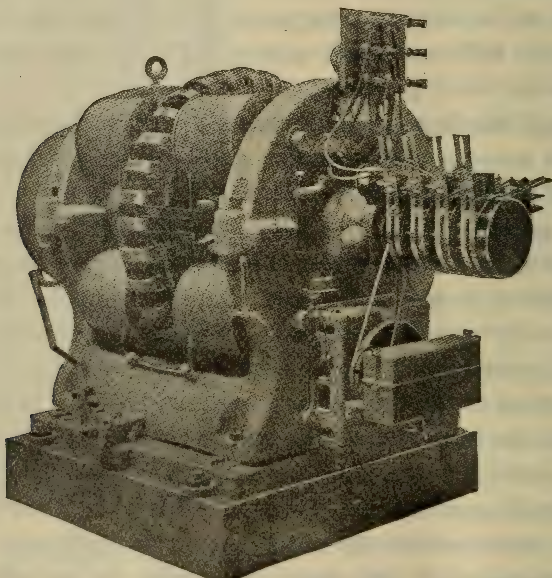


Fig. 102. Multipolar Brush arc generator.

Unless the generator can be set up on a substantial floor a foundation of masonry must be built.

In whatever manner the bed-plate is mounted, the greatest care must be taken to have a thorough and permanent insulation from earth.

Four short bolts pass through the generator frame and are used to hold the machine in position on its bed-plate. They are inserted in the slots in the iron base-plate and provided with square nuts at their lower ends.

The lower half of the frame is first placed in position on the bed-plate and bolted down.

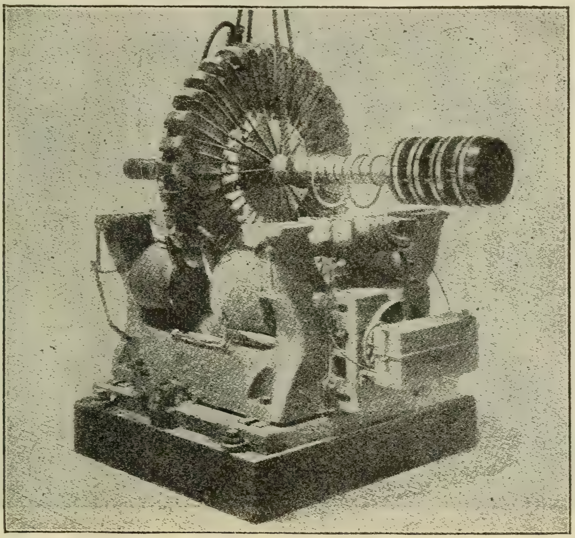


Fig. 103. Method of suspending armature.

The lower halves of the bearing boxes should be removed and the oil chambers thoroughly cleaned and filled with a good quality of stringy oil, to the height indicated by the mark on the oil gauge. The lower halves of the bearing boxes may then be replaced.

The proper method of suspending the armature is shown in Fig. 103.

When handling the magnet yokes, a rope sling should be

used, as shown in the illustration, Fig. 104. The bolts should be inserted as shown, and as the yoke is lowered, these will act as a guide and drop it into its proper place. The frame bolts must be

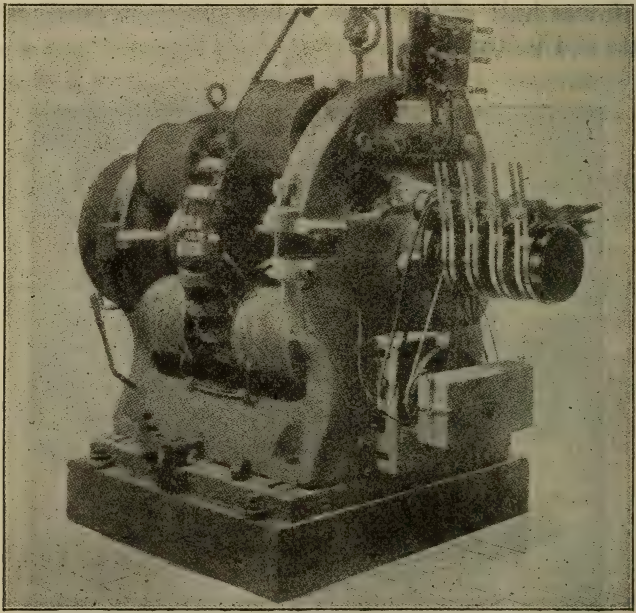


Fig. 104. Method of handling the magnet yoke.

screwed up especially tight, as any movement of the yokes while the machinery is running will ruin the armature.

The brush-holder yoke and the regulator rocker arm should be put in place with a little oil on the bearing seats to insure freedom of movement through the entire range.

SETTING THE BRUSHES.

A pressure brush should always be used over the under brush, as it improves the running of the commutator and secures a bet-

ter contact on the segment. The brushes should be set $5\frac{1}{8}''$ from the front side of the brass brush-holder.

In setting the brushes, commence with the inner pair and set one brush about $5\frac{1}{8}''$ from the holder to the tip of brush, then rotate the rocker or armature until the tip of the brush is exactly in line with the end of a copper segment as shown in Fig. 105. The other brush should be set on the corresponding segment 90° removed, but if the length of the brush from the holder is less than $5\frac{1}{8}''$, move both brushes forward until the length of the shorter brush from the holder is $5\frac{1}{8}''$. Now set the two extreme outer brushes in the same manner, clamping firmly in position, and by using a straight edge or steel rule, all the brushes can be set in exactly the same line and firmly secured. The spark on one of the six brushes may be a trifle longer than on the others. In this case, move the brush forward a trifle so as to make the sparks on the six brushes about the same length. Equality in the spark lengths is not essential, but it gives at a glance an indication of the running condition of the machine.

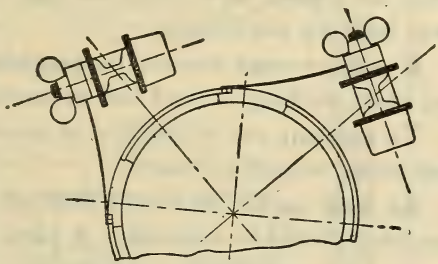


Fig. 105. Position of brushes.

Fig. 106.

Brushes should not bear on the commutator as illustrated in Fig. 106; they will tend to drop into the commutator slots and pound the copper tip of the wood block. If the bearing is too far from the end as in Fig. 107, the point of the brush is lifted from the leaving end of the segment, causing sparking.

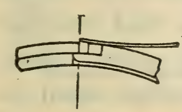


Fig. 107.

Fig. 108 shows correct setting.

Fig. 108.

CARE OF COMMUTATOR.

If the commutator needs lubrication, oil it very sparingly. Once or twice during a run is ample. If the oil has a tendency to blacken the commutator instead of making it bright, wipe the commutator with a dry cloth.

The machine, of course, generates high potential, and the cloth, or whatever is used to oil the commutator, should, therefore, be placed on a stick so that the hand is not placed in any way between the brushes.

A rubber mat should be provided for the attendant to stand on when working around the commutator and brushes.

To prevent any possibility of shock, all switches on the terminal board should be closed.

As soon as the current is shut off from the machine, the commutator should be cleaned. A piece of very fine sandpaper held against the commutator under a strip of wood for about a minute before the machine is stopped, will scour the commutator sufficiently. The brushes need not be removed. Never use a file, emery cloth, or crocus, on the commutator. New blocks will sometimes cause flashing, due to the presence of sap in the wood.

CONNECTIONS OF MULTIPOLAR BRUSH ARC GENERATORS.

Connections of Multipolar Brush Arc Generators are shown in Diagram Fig. 109. The current enters the field from the negative side of the circuit and takes the following course: Spool 1, to 2, to 7, to 8, to 5, to 6, to 3, to 4, to terminal board, to commutator. The field current is in the same direction for clockwise and counter clockwise machines.

SINGLE CIRCUIT, CLOCKWISE ROTATION
WITH FORM I REGULATOR

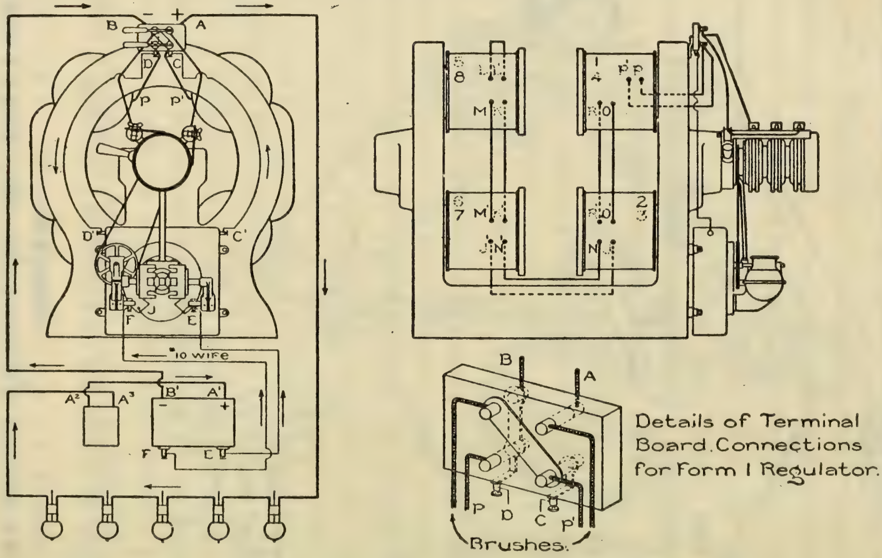


Fig. 109.

CONNECTIONS OF Nos. 8½, 9, 10 AND 11 BRUSH ARC GENERATORS

The current of the Brush Arc Machine is automatically maintained constant by a regulator of one of the forms described on the following pages.

FORM 1 REGULATOR FOR MULTIPOLAR BRUSH ARC GENERATORS.

The Form 1 Regulator is placed on the frame of the machine

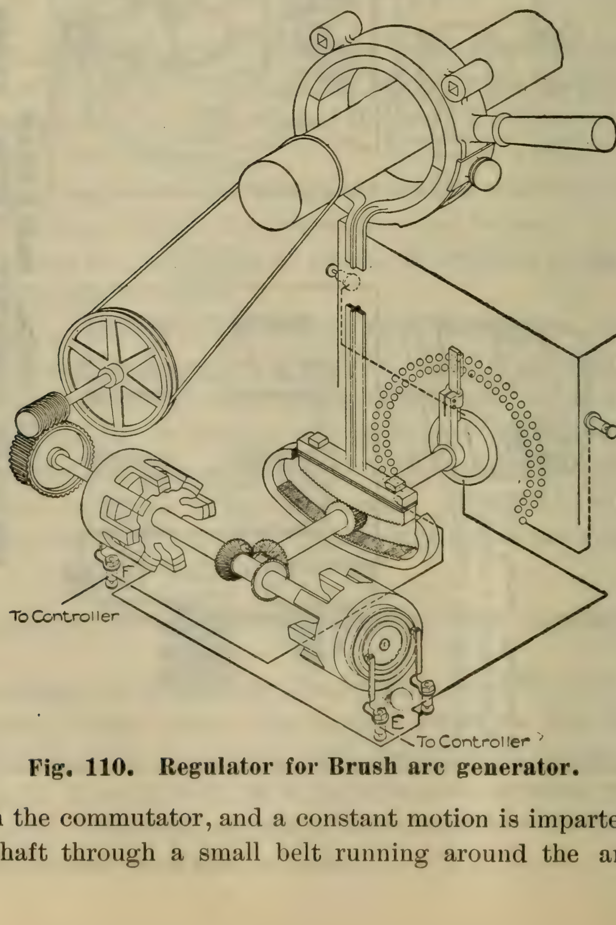


Fig. 110. Regulator for Brush arc generator.

beneath the commutator, and a constant motion is imparted to its main shaft through a small belt running around the armature

shaft. (See Fig. 110.) By means of magnetic clutches and bevel gears, a pinion shaft is rotated, which moves the rack and the rocker arm and so shifts the brushes on the commutator; at the same time the rheostat arm is moved over the contacts to cut resistances in or out of the shunt around the field circuit.

The current for the magnetic clutches is regulated by the controller.

The controller consists principally of two magnets which are energized by the main current and act when the current is too high or too low, by sending a small current to one of the clutches.

If the controller is out of adjustment and fails to keep the current normal, do not try to adjust the tensions of both armatures at the same time. The left-hand spool *I* (see Diagram Fig. 111) may not take hold quickly enough, or the spool *F* may take hold too quickly. To make the adjustment, screw up the adjusting button *K* on the right-hand spool, increasing the tension. This will have a tendency to let the current fall much lower before the armature comes in contact with *H*, to cause the current to increase. By simply tapping the armature *G* quickly with a pencil or piece of wood, forcing it down with its contact, and at the same time watching the ammeter, the current may be brought up to 6.8 amperes if 6.6 amperes is normal, or 9.8 if 9.6 is normal. With the current at 6.8 amperes, which is .2 amperes high, the adjusting button *L* should be turned to increase the tension on this spring until the armature *M* comes in contact with contact *N*, which will force current down through *O*. The clutch which pulls the brushes forward and rocks the rheostat back for less current will thus be energized. Repeat this adjustment two or three times, but do not touch the adjusting button *K*; adjust *L* until it is just right.

At the side of the armature *M* a little wedge is screwed in by means of an adjusting button, and increases or decreases the leverage on this armature. See that this wedge is fairly well in

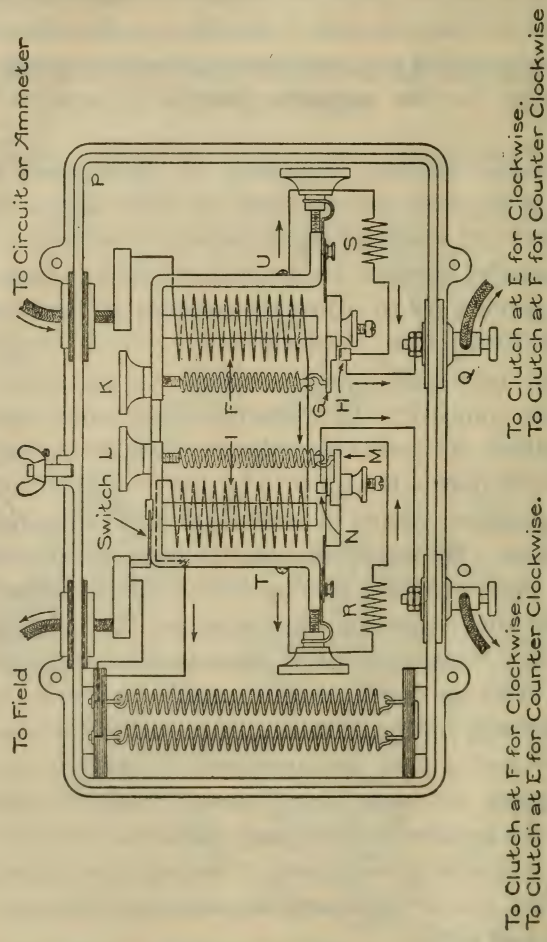


Fig. 111. Connections of Brush controller.

between the core or frame of the spool and the spring of the armature. The armature M may have to be taken out and the spring slightly bent. It is advisable to have the screw which passes through the adjuster button L about half way in, to allow an equal distance up and down for adjusting this lighter spring after the wedge-shaped piece is in the right position to give the necessary tension on the spring which is fastened to the armature M.

Having adjusted the spool I so that the current will not rise above 6.8 (or 9.8) amperes, move the armature M up to contact N with a pencil or piece of wood, causing the current to be reduced to about 6.2 (or 9.2). After the current settles at this point, decrease the tension on the spring which is fastened to armature G, allowing this armature to fall down to contact H. Current will then flow through Q, which will rock the brushes back and also move the rheostat arm for more current. As the spool I has been adjusted for 6.8 (or 9.8) amperes, the current cannot rise above that amount, no matter how the spool F is adjusted.

The Two small shunt coils, R and S, are connected around the two contacts simply to decrease the spark between the silver and platinum contacts. If they should become short circuited in any way, so that their resistance becomes diminished, sufficient current may pass through either of them to operate the regulator. If unable to locate the trouble, disconnect these coils at points T and U, when a thorough examination can be readily made. M and G need not move more than just enough to open the contact — $\frac{1}{32}''$ is ample.

For electric elevators, their care and management see page 716.

STARTING THE MULTIPOLAR BRUSH ARC GENERATOR WITH FORM 1 REGULATOR.

In starting, the lower switch, which short circuits the field, should be opened last.

The switch in the left-hand corner of the controller (Diagram Fig. 111) cuts out the two resistance wires which are used to force the current through wires O and Q to the clutches. Open this switch. Unclasp the brush rocker from the rheostat rocker. Move the brushes by hand to give the proper spark, allowing the rheostat arm to be moved by the controller. After the switches are opened, the rhoestat arm will go clear around to a full load position, and then, the controller takes hold and brings the arm back. In the meantime, rock the brushes forward or backward and keep the spark about the proper length, say $\frac{1}{8}''$ at full load to $\frac{3}{8}''$ on short circuit. Gradually the rheostat arm will settle, the spark will become constant, and the machine will give its proper current. Then clamp the rocker and rheostat arm together and let the machine regulate itself.

FORM 2 REGULATOR FOR MULTIPOLAR BRUSH ARC GENERATORS.

The connections of the Form 2 Regulator are shown in Fig. 113. The regulator performs two operations; sweeps a set of contacts, throwing more or less resistance in shunt with the field circuit, and at the same time, rocks the brushes so that the spark is kept at proper length, varying at from $\frac{1}{8}''$ at full load to $\frac{3}{8}''$ on short circuit.

A small belt runs over the armature shaft M and drives the

rotary oil pump P. The pump draws the oil from the containing case and forces it through passages to the valve T.

The ports overlap this valve so that the oil may flow through when the valve is in its central position. The valve is controlled by the electromagnet F (Fig. 113) which actuates the armature U

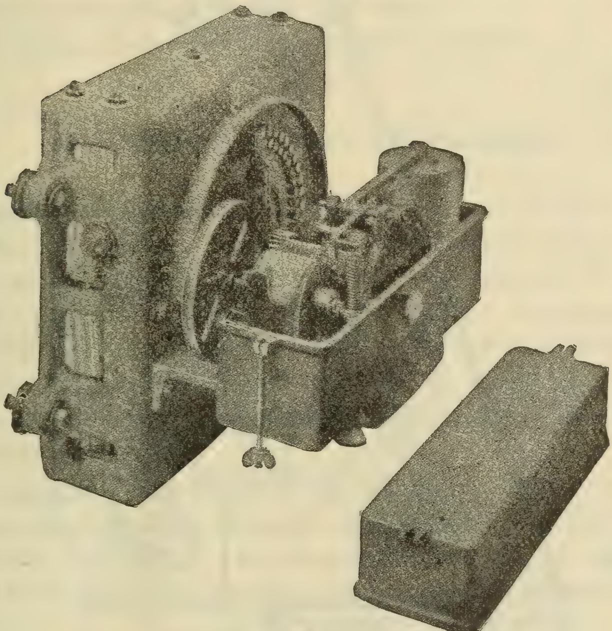


Fig. 112. Form 2 regulator.

and the lever H. The pull on armature U varies with the strength of the current which excites F. The opposite end of the lever H is attached to spring G, which is adjusted by the screw nut R so as to hold the valve in central position when normal current is flowing through the controlling magnet.

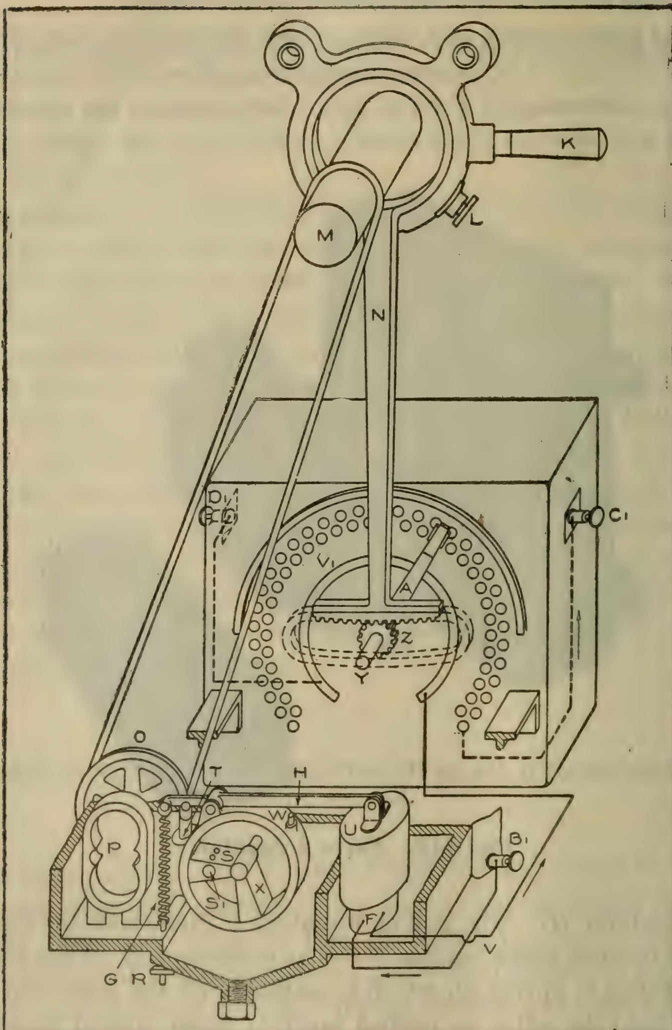


Fig. 113. Form 2 regulator for Brush arc generator.

If the current is too strong, it pulls down the armature *U*, raising the valve, throwing more oil on the upper side of the circular piston head *S*, and allowing the oil to run out from the lower side, thus forcing the piston *X* around clockwise, lowering the current by moving the contact arm so as to shunt more current from the fields, at the same time moving the brushes forward until the current returns to its normal value.

If the current is too low the operation is reversed.

ADJUSTMENT OF FORM 2 REGULATOR.

To raise the current, turn the hard rubber nut *R* (Fig. 113) to the right. If the current is too high, turn the nut to the left.

The limits between which the regulator operates are determined by the number of turns in the spring *G*. If the spring *G* is stiffened by cutting off some of the turns and stretching it out, the limits of regulation will be wider. If the spring has a greater number of turns, it will regulate within narrower limits, but be more liable to "pump."

The regulator may be caused to operate quickly in one direction and slowly in the reverse direction by changing the position of the stops on lever *H*. By raising the stop on the right-hand side of lever *H*, the movement on increasing the current will be retarded.

STARTING THE MULTIPOLAR BRUSH ARC GENERATOR WITH FORM 2 REGULATOR.

Before starting the machine, the oil box of the regulator should be filled with a light spindle or dynamo oil nearly up to the shaft which carries the contact arm, etc.

If the pump fails to start promptly, it may be started by shift-

ing the brushes backward and forward, and moving the contact arm.

In a newly installed machine, the oil should be changed at least once a week.

Having correctly adjusted the regulator for the desired current, as previously described, the starting valve handle S^1 (Fig. 113) should be turned counter clockwise when the machine is running without load. This handle operates a valve which connects both sides of the circular cylinder, thereby giving a free flow of oil between the two sides, and preventing the operation of the piston and relieving the pump from any undue load.

To put the machine in operation, the valve should be gradually thrown around clockwise, cutting off the flow of oil from the two sides of the cylinder after the switches have been opened. This valve may also be used to throw the regulator out of operation if desired.

FORM 3 REGULATOR FOR MULTIPOLAR BRUSH ARC GENERATORS.

In the Form 3 regulator a belt from the armature shaft runs a small countershaft with crank attached. A rocking or reciprocating motion is thus imparted to the main lever, on which are pivoted two self-adjusting clutch jaws or grips. When the current is normal, the clutches are held stationary, but as the current varies, either above or below normal, the clutch on one side is dropped so that it will grip the clutch disk, and the mechanism revolves in the proper direction to restore the current.

With slight variations of the current, the regulator contact arm is moved forward or backward very slowly; while, with greater variations, caused by any considerable number of lamps being cut in or out, the movement is increased and the normal point or position more quickly reached.

For starting Brush Arc Generators with Form 3 Regulators, see general directions under Form 1 Regulator.

FORM 4 REGULATOR FOR MULTIPOLAR BRUSH ARC GENERATORS.

The Form 4 Regulator is similar to the Form 3; the countershaft and rocking lever are identical, but instead of using clutches, the lever operates two pawls, which engage in ratchet wheels. The pawls are not in contact when the current is normal, but are thrown in to move the arm to either right or left as the current varies and the regulator is called upon to shunt more or less current from the field.

AMMETER.

A reliable ammeter should always be connected in the circuit of an arc generator, so that the exact current may be read at a glance. It should be connected into the negative side of the line where the circuit leaves the regulator.

INSTRUCTIONS FOR INSTALLING AND OPERATING IMPROVED BRUSH ARC LAMPS.

Suspension. — One of three methods of suspension may be used for Brush Arc Lamps. If chimney suspension, which is the most common, is adopted, the wire, cable or rope used to suspend the lamp must be carefully insulated from the chimney. For this purpose a porcelain insulator should be inserted between the support and the lamp.

Hook suspensions may be used to advantage in some places, but greater care must be taken to insulate the supporting wires from any conductors, as the hooks form the terminals of the lamp.

The most convenient arrangement for indoor use is to suspend the lamp from a hanger board. The porcelain base of the hanger board prevents short circuits or grounds.

The lamps run nominally on circuits of 6.6 amperes for 1200 candle-power and 9.6 amperes for 2000 candle-power. In case it is necessary to run a lamp on a circuit differing from the standard, the lamp may be adjusted by moving the contact on the adjuster. This will compensate for about one ampere either way from normal and is set in about the middle position when the lamp is shipped.

Permanent adjustment for special circuits of variation greater than one ampere from standard is made by filing the soft iron armature. The clutch should be so adjusted that the center of the armature is $\frac{13}{16}''$ above the plate when the trip on the first rod is touching the bushing and $\frac{11}{16}''$ when the trip on the second rod is in a similar position. A small gauge is convenient for adjusting the clutch. The position of the trip of the clutch determines the feeding point of the lamp.

After thoroughly repairing and cleaning the lamp, it should be run a short time before installing. Lamps should not be tested in an exposed place, as a strong draft of air will cause unpleasant hissing, which may be mistaken for some internal trouble.

Lamps should not hiss or flame if good carbons are used. A voltmeter should always be used when adjusting or testing.

Connecting. — The lamp terminal hooks are marked *P* (positive) and *N* (negative), and should be connected into circuit accordingly. See Fig. 114.

The carbons should rest in contact when the lamp is cut out. When the switch is opened, part of the current from the positive terminal hook (*P*) goes through the adjuster to the yoke, and thence through the carbon rod and carbons to the negative terminal hook (*N*). The remainder of the current goes to the cut-out block, but, as the cut-out is closed at first, the current crosses

over through the cut-out bar to the starting resistance, and so to the negative side of the lamp. A part of it, however, is shunted at the cut-out block through the coarse wire of the magnets, and so to the upper carbon rod and carbons and out. This shunted current energizes the magnets and so raises the armature which opens the cut-out and at the same time establishes the arc by separating the carbons.

The fine wire winding is connected in the opposite direction from the coarse winding, and its attraction is therefore opposite. When the arc increases in length, its resistance increases, and consequently, the current in the fine wire is increased. The attraction of the coarse wire winding is, therefore, partly overcome and the armature begins to fall. As it falls, the arc is shortened and the current in the fine wire decreases. The mechanism feeds the carbons and regulates the arc so gradually that a perfectly steady arc is maintained.

The fine wire of the magnets is connected in series with the winding of a small auxiliary cut-out magnet at the top of the mechanism.

This magnet, which also has a supplementary coarse winding, does not raise its armature unless the voltage at the arc increases to 70 volts. The two windings connect at the inside terminal on the lower side of the auxiliary cut-out magnet, and the current from the fine wire of the main magnets passes through both windings and then to the cut-out block and so to the starting resistance and out.

If the main current through the carbon is interrupted (as by breaking of the carbons), the whole current of the lamp passes through the fine wire circuit. Before this excessive current has time to overheat the fine wire circuit, it energizes the auxiliary cut-out magnet and closes a circuit directly across the lamp through the coarse wire on the auxiliary cut-out to the main cut-out block, and thence to the negative terminal.

The auxiliary cut-out operates instantly and prevents any dan-

ger to the magnets during the short period required for the main armature to drop and throw in the main cut-out. When the main

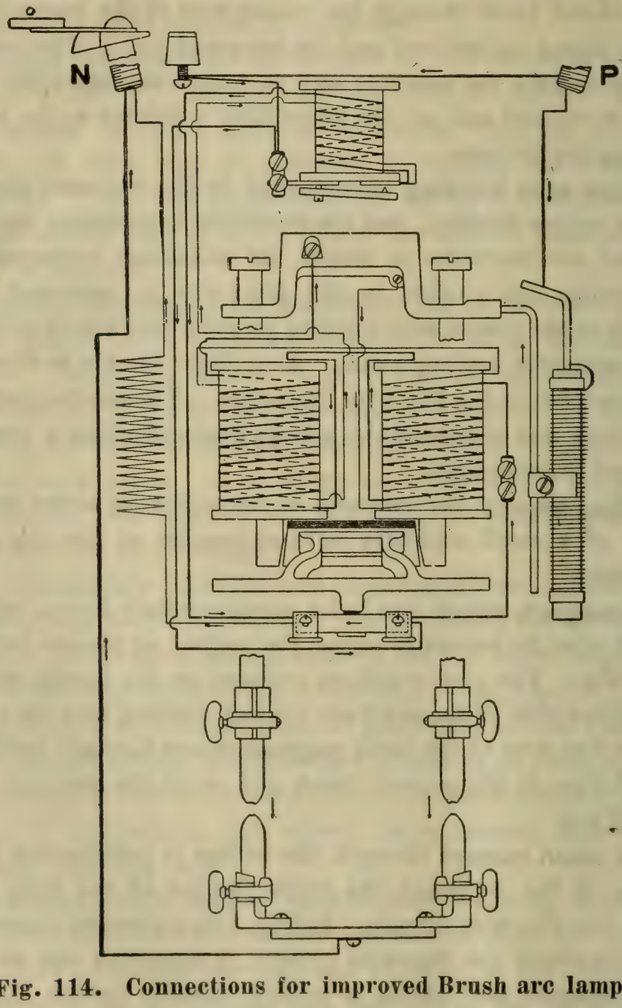


Fig. 114. Connections for improved Brush arc lamps.

cut-out operates, the armature of the auxiliary cut-out fails, because there is not sufficient current in that circuit to energize the magnet.

The voltage at which the auxiliary cut-out magnet operates depends on the position of its armature, which is regulated by the screw securing the armature in position. It should not be adjusted to operate at less than 70 volts.

The carbons should be solid and of uniform quality. For the best results, the upper carbon should be $12'' \times \frac{7}{16}''$, and the lower $7'' \times \frac{7}{16}''$. The stub of the upper carbon may then be used in the lower holder when retrimming.

At each trimming the rod should be carefully wiped with clean cotton waste. It should never be pushed up into the lamp in a dirty condition.

In order to remove the carbon rod or examine the mechanism, the jacket must be lowered by pressing a spring clip on its under side.

The carbon rod may be unscrewed and removed by a small screw-driver or small strip of metal inserted in the slot cut in the rod cap. The cap will remain in the hole through the yoke when the rod is taken out.

The lamp must never be left burning with the jacket off, nor be allowed to hang with the mechanism exposed to the weather.

PERSONAL SAFETY.

Never allow the body to form part of a circuit. While handling a conductor, a second contact may be made accidentally through the feet, hands, knees or other part of the body in some peculiar and unexpected manner. For example, men have been killed because they touched a "live" wire while standing or sitting upon a conducting body.

Rubber gloves or rubber shoes, or both, should be used in handling circuits of over 500 volts. The safest plan is not to touch any conductor while the current is on, and it should be remembered that the current may be present when not expected, due to an accidental contact with some other wire or to a change

of connections. Tools with insulated handles, or a dry stick of wood, should be used instead of the bare hand.

The rule to use only one hand when handling dangerous electrical conductors or apparatus is a very good one, because it avoids the chance, which is very great, of making contacts with both hands and getting the full current right through the body. This rule is often made still more definite by saying, "Keep one hand in your pocket," in order to make sure not to use it. The above precautions are often totally disregarded, particularly by those who have become careless by familiarity with dangerous currents. The result of this has been that almost all the persons accidentally killed by electricity have been experienced electric linemen or stationmen.

TABLE SHOWING RELATIVE RESISTANCE OF METALS AT TEMPERATURE OF 70 DEGREES F.

NAME OF METAL.	Resistance in Ohms of Wire 100 ft. long and one-thousandth of an inch in diameter.	
		Remarks.
Silver	965	The resistance of arc light carbons is given for comparison and, as will be noticed, it is about 4000 times as great as that of silver.
Copper	1,030	
Gold	1,328	
Aluminum	1,900	
Zinc	3,600	
Platinum	5,700	
Iron, Wrought	6,400	
Nickel	7,500	To obtain the resistance of 100 ft. of wire of any size, divide the figures in this table by the square of the diameter of the wire in thousandths of an inch.
Tin	8,500	
German Silver	12,600	
Lead	12,700	
Antimony	23,000	
Manganese Steel	42,000	
Mercury	57,700	
Arc Light Carbon	3,792,000	

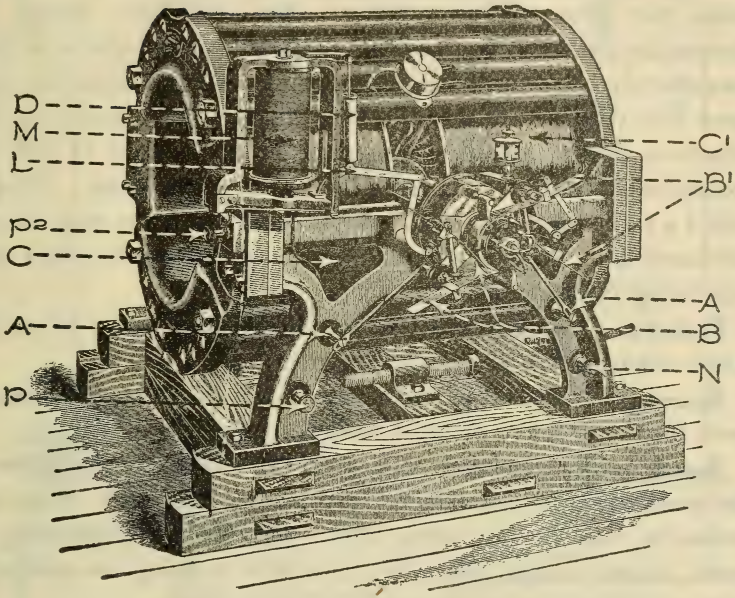


Fig. 115. The Thomson-Houston standard arc dynamo arranged for right-hand rotation.

CHAPTER X.

INSTALLATION OF ARC DYNAMOS.

Location and mounting. — The generator should be located in a cool, dry room, free from dust, metal chips or flying particles of any sort. Space should be allowed around the machine to give ample room for reaching all parts of it, particularly the commutator. The generator should be set upon a firm founda-

tion of well-seasoned wood, and should be mounted upon a sliding bed-plate, so that the belt can be tightened or loosened while the generator is running. The generator should be thoroughly insulated from earth. The sliding bed-plates as now manufactured are designed to provide perfect insulation, and meet this requirement fully. The direction of rotation of the armature in the standard generator is from right to left, or counter-clockwise, as seen when facing the commutator. This is called a right-hand machine. Right-hand machines may be run left-handed by replacing certain parts of the brush-holder and regulating mechanism.

Pulleys. — The generator is provided with a pulley of proper size to transmit the power demanded.

Bearings. — The oil in the reservoir should be renewed once a week for the first two or three weeks.

Speed. — The generator should be run as nearly as possible at the speed given by the maker. An increase of speed, if not too excessive, will do no harm, but a considerable diminution in speed below normal, when the generator is doing its maximum work, is liable to cause unsteadiness in the lights.

The automatic regulator will adjust perfectly for fluctuations in speed near or above normal, unless the fluctuations are extremely sudden, as in the case of slipping of the belt.

Belts. — The belt should be about half an inch narrower than the face of the pulley. An endless belt is desirable.

Brushes. — When the generator is in position the brushes or strips of copper B B, B^1 B^1 (see Fig. 117), are placed on the machine in the manner shown. All four brushes should be set exactly to the gauges sent with each machine, so that they press with sufficient force on the surface of the commutator to insure good contact at all times.

The length of the gauge is such that the brushes project a little past the center of the commutator, as shown in Fig. 119, to

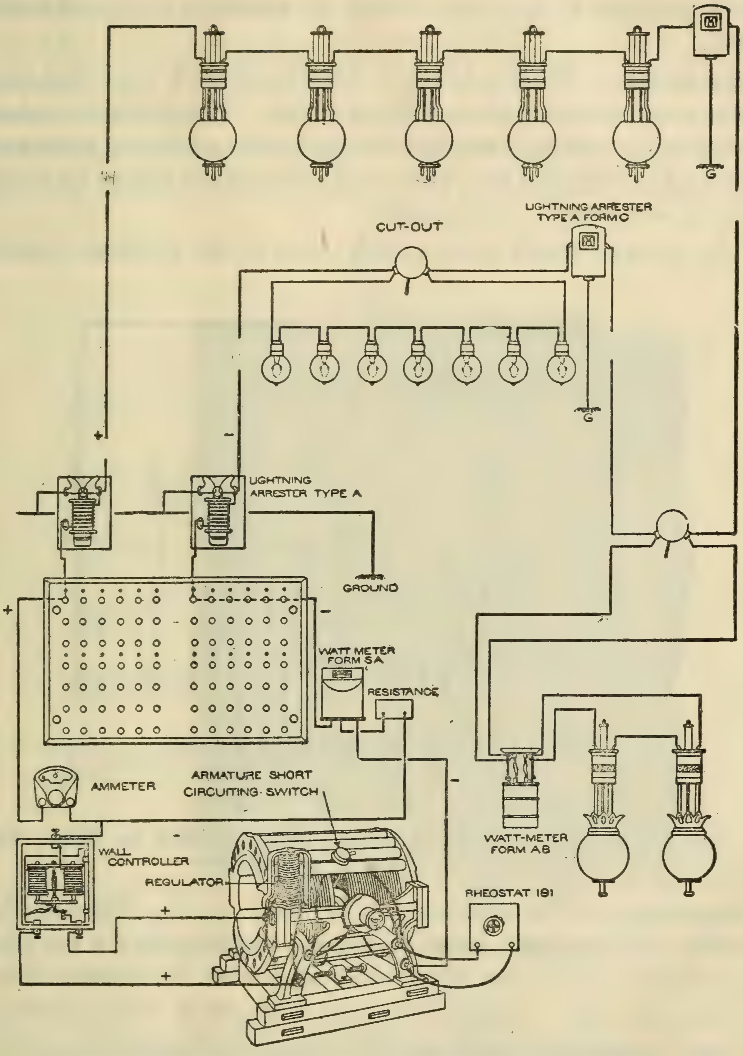


Fig. 116. Connections for arc lighting system.

avoid catching in the slots should the armature be turned backward.

Air Blast. — The air blast or blower plays an important part in the successful operation of the machine. The air blast requires no attention, except that it should be kept scrupulously clean and well oiled. Only the best quality of mineral oil should be used. Poor oil will always cause trouble.

The screens which cover the air inlets on the air blast, should

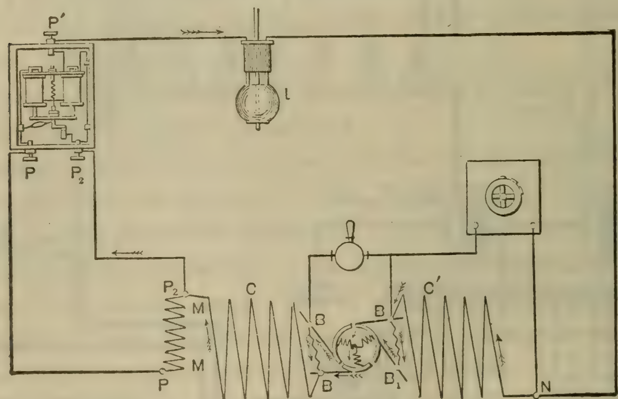


Fig. 117. Connections for rheostat.

be kept clean and free from dust. They should be taken out about once a month and cleaned in kerosene oil.

Regulator. — The regulator is fastened to the frame of the machine by two short bolts, as shown in Fig. 116. On the left-hand machine, *i. e.*, one which runs clockwise, its position is on the opposite side. Before filling the dash-pot *D* with glycerine, see that the regulator lever and its connections, brush yokes, etc., are free in every joint, and that the lever *L* can move freely up and down. Then fill the dash-pot *D* with concentrated glycerine.

The long wire from the regulator magnet M, is connected with the left-hand binding post P of the machine, and the short wire with the post P^2 on the side of the machine. The inside wire of the field magnet, or that leaving the iron flange of the left-hand field should be connected into the post P^2 also, as shown in Fig. 116. The electric circuit (see Fig. 117), should be complete from

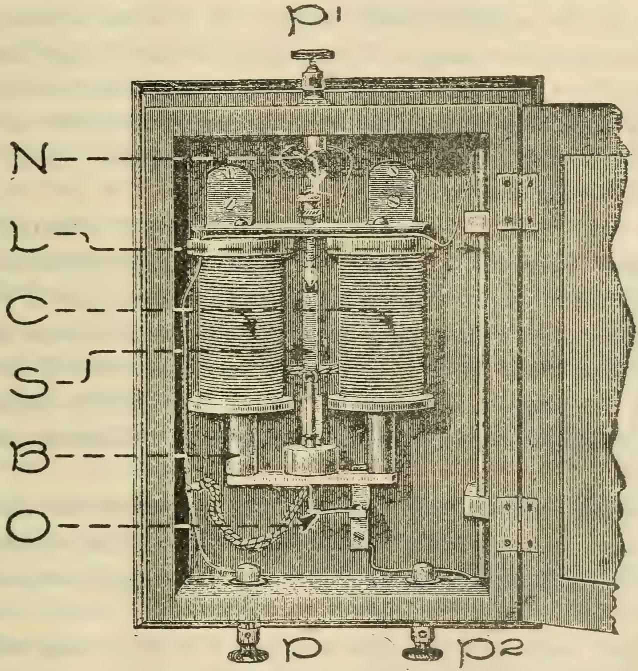


Fig. 118. Controller for arc dynamo.

P^1 on the controller magnet, through the lamps to post N on the machine, through the right-hand field magnet C^1, to the brushes B^1 B^1, through the commutator and armature to the brushes

B B, through the left-hand field C, to posts P^2 and P, thence to posts P^2 and P on the controller magnet, through the controller magnet to P^1. The current passes in the direction indicated by the arrows.

Controller. — The controller magnet (see Fig. 118) is to be fastened securely by screws to the wall or some rigid upright support, taking care to have it perfectly plumb. It is connected to the machine in the manner shown in Fig. 117, *i. e.*, the binding post P^2 on the controller magnet, is connected to the binding post P^2 (see Fig. 2) on the end of the machine; and likewise, the post P on the controller to the post P on the leg of the machine; the post P^1 forms the positive terminal from which the circuit is to run to the lamps and back to N.

Great care should be taken to see that wires P P and P^2 P^2 are fastened securely in place; for if the connection between P and P should be impaired or broken, the regular magnet M would be thrown out of action, thus throwing on the full power of the machine, and if the wires P^2 P^2 should become loosened, the full power of the magnet M would be thrown on, and the regulator lever L, rising in consequence, would greatly weaken or put out the lights.

The wires leading from the controller magnet to the machine should have an extra heavy insulation. Care should be taken in putting up the controller magnet that the following directions are followed: —

(1) The cores B of the axial magnets C C must hang exactly in the center, and be free to move up and down.

(2) The screws fastening the yoke or tie pieces to the two cores must not become loosened.

(3) The contacts O must be firmly closed when the cores are not attracted by the coils C C, which is the case, of course, when no current is being generated by the machine, and when the cores

are lifted, the contacts must open from $\frac{1}{64}''$ to $\frac{1}{32}''$; a greater opening than $\frac{1}{32}''$ has the effect of lengthening the time of action of the regulator magnet. This tends to render the current unsteady, and in case of a very weak dashpot or short circuit, might cause flashing. If this adjustment is not properly made there will be destructive sparking on a small portion of the contact surfaces.

(4) All connections must be perfectly secure.

(5) The check-nuts *N* must be tight.

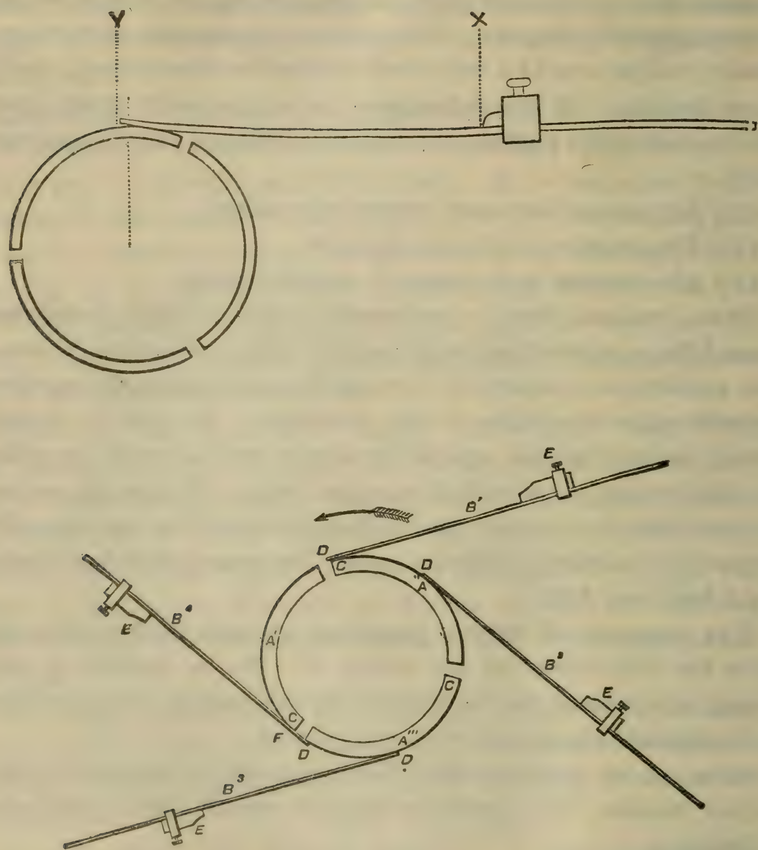
(6) The carbons in the tubes *L* must be whole.

These carbons form a permanent shunt of high resistance around the regulator magnet *M*, and if broken will cause destructive sparking at contacts *O*, burning them and seriously interfering with close regulation of the generator. In case a carbon should become broken, temporary repairs may be made by splicing the broken piece with fine copper wire. To keep the action of the controller perfect, the contacts *O* should be occasionally cleaned by inserting a folded piece of fine emery cloth and drawing it back and forth.

The amount of current generated by each machine depends upon the adjustment of the spring *S*. If the tension of this spring is increased, the current will be diminished; if the tension is diminished, the current will be increased.

Once set up and in perfect working condition, adjusted to the proper current, the controller magnet should rarely need any adjustment.

Testing arc light dynamos. — The commutator should fit the shaft snugly, but be sufficiently free to turn easily on the shaft. Be very careful to put the short brush-holders on the outer yoke, and the long brush-holders on the inner yoke. Also see that the long binding post, attached to the sliding connection, is on the lower left-hand brush-holder, and the short post on the lower right-



A — Commutator Segments.
B^2 B^4 } Primary Brushes.
B^1 B^3 } Secondary Brushes.

C—Forward Point of Segment.
D—Point of Brush.
E—Brush-holders.
F—Point of Contact.

Figs. 119 and 120. Showing position of brushes.

hand brush-holder. Always set the brush-holders to the proper angle by the brush-holder gauge. First tighten up the brush-holders and then turn them to the correct position by means of a piece of steel wire passed through the holes. Then permanently tighten up the brush-holders very firmly, trying them with the gauge to see that they are the same distance from the commutator. Always be careful to get the brushes exactly straight and flat before clamping them to the brush-holders, and always set them to the exact length of the brush gauge.

Setting the cut-out. — After the brushes are in position, the cut-out must be set. This is done by turning the commutator on the shaft in the direction of rotation (if the commutator is set in position the whole armature must be revolved), until any two segments are just touching the primary brush on that side, as segments A' and A''' touch brush B^1 in Fig. 120. Under these conditions brush B^4 should be at the left-hand edge of upper segment. Then turn commutator until the same two segments are just touching brush B^2, when the end of brush B^3 should just come to the right-hand edge of the lower segment. If the secondary brush projects beyond the edge of the segment the regulator arm should be bent down; if it does not come to the edge of the segment the arm should be bent up.

Care must be taken that the regulator armature is down on the stop when the cut-out is being set.

Always try the cut-out on both primary brushes. If it does not come the same on both, turn one over. If the brush-holders are correctly set by the gauge, there should be no trouble in getting the cut-out set properly after one or two trials.

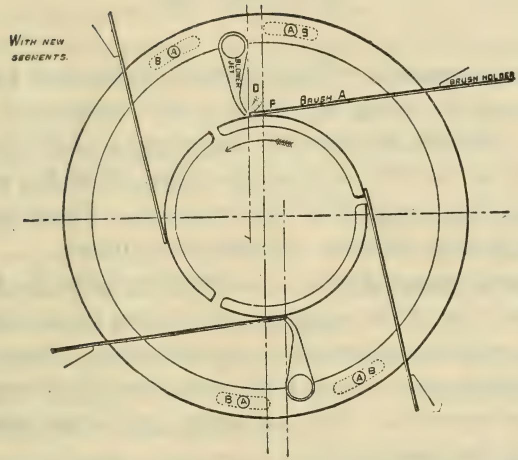
The distance from the tip of the brush, which is on top, to the left-hand edge of No. 2 segment on a right-hand machine, or to the right-hand edge of No. 3 segment in a left-hand machine, is called the lead, and should be made to correspond to the following table: —

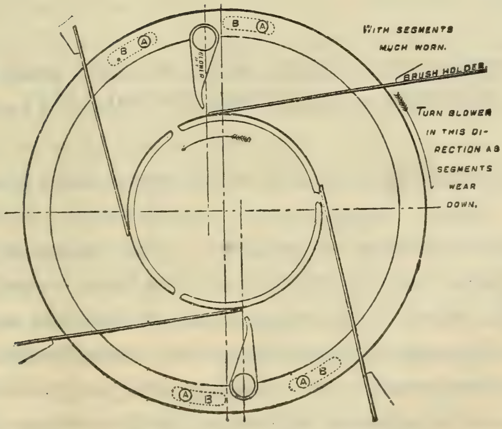
TABLE OF LEADS.

DRUM ARMATURES.	RING ARMATURES.
C^{12} $\frac{1}{4}''$ positive.	K^{12} $\frac{3}{16}''$ positive.
C^{2} $\frac{3}{8}''$ "	K^{2} $\frac{1}{8}''$ "
E^{12} $\frac{7}{16}''$ "	M^{12} $\frac{1}{4}''$ negative.
E^{2} $\frac{1}{4}''$ "	M^{2} $\frac{1}{2}''$ "
H^{12} $\frac{1}{4}''$ "	LD^{12} $\frac{1}{4}''$ positive.
H^{2} $\frac{1}{4}''$ "	LD^{2} $\frac{1}{8}''$ "
	MD^{12} $\frac{13}{32}''$ "
	MD^{2} $\frac{13}{32}''$ "

Place the screws in the binding posts at the lower ends of the sliding connections and put on the dash-pot connections between the brushes, with the heads of the connecting screws outward. In every case the barrel part of the dash-pot is connected to the top brush-holder, and plunger part to the bottom brush-holder. See that the field and regulator wires are connected and that all connections are securely made. When all connections have been made, make a careful examination of screws, joints, and all moving parts. They must be free from stickiness, and not bind in any position.

To determine when the machine is under full load, notice the position of the regular armature, which should be within $\frac{1}{8}''$ of the stop. At full load the normal length of the spark on the commutator should be about $\frac{3}{16}''$. If it is less than this, shut down the machine and move the commutator forward, or in direction of rotation until the spark is of the desired length. If the spark is too long, move the commutator back the proper amount.





Lift Regulator as high as possible.

Figs. 121 and 122. Position of air blasts on LD and MD dynamos.

DIRECTIONS FOR SETTING THE AIR BLAST JETS ON LD AND MD DYNAMOS.

With new segments.— Loosen bolts *A–A–A–A* and turn the air-blast so as to bring the bolts in the centers of the slots *B–B–B–B*. Set the brushes by the gauge. Lift the regulator lever as high as possible and set the point *D* of the air blast jet $\frac{1}{32}''$ in front of the point *P* of the brush *A*. Place the lower jet in the same relative position with the lower brush.

As segments wear down.— Loosen the bolts *A–A–A–A* and follow up the wear of the segments by turning the air blast against direction or rotation of armature as indicated. Turn the point of the jet downward, so as to blow more directly through the slot between the segments. Set the lower jet in the same relative position with the lower brush.

SOME TROUBLES WHICH MAY BE MET AND THEIR CAUSES — REVERSAL OF POLARITY.

Cases are frequently reported where generators, from lightning discharges, wrong plugging on switch-board, or some other reason, suffer a reversal of polarity. The effect of reversal is that the lamps in circuit with the machine burn "upside down;" which has the effect of throwing much of the light up instead of down, and with some carbons the arc will flame badly. This can be remedied temporarily, by changing the plugs on the switch-board, so that the current will enter the line where ordinarily it returns.

Occasion should be taken, however, as soon thereafter as

possible, to properly magnetize the fields so that they will be the right polarity, as follows: —

Close the armature short circuiting switch on the frame of the machine and run a loop from some other arc generator which happens to be in operation. Connect the positive side of this loop to the lower binding post N on the right leg of the machine, and the negative side of the binding post P^2 on the end of the frame under the regulator. Then open the armature short circuiting switch on the second generator. A very few seconds will suffice to correctly polarize the first machine.

To detect a short circuit in the field, make all adjustments as if working under normal conditions, then run the machine at the proper speed on a dead short circuit. If there is no short circuit in the field, the armature of the regulator will be drawn up hard against the bottom of the magnet, but if there is a short circuit in the field the armature will drop more or less according to the amount of field wire cut out of circuit.

To find out which half of the field is affected, close the field switch and remove the regular wire from the Post P^2, Fig. 115, then connect posts P^2 and N to some source of direct current, as a 110-volt exciter, and with a volt-meter measure the drop in voltage between posts N and A^1 and between A and P^2. The drop should be very nearly the same in both cases if the winding is perfect, but the drop will be less across that field which is short circuited.

Another trouble which is liable to be met is flashing. When a generator flashes an arc is drawn around the commutator from one brush to the other, which soon short circuits the armature, putting out the lights. This arc is usually broken very quickly, but the flashing may be repeated at frequent intervals. There are several causes of flashing, such as overload, low speed, stickiness in the regulating mechanism, short circuit in the field, commutator not in proper position, or a dash-pot which is too stiff or

too loose. If a machine flashes when running under proper load and at proper speed, see that there is no stiffness in the regulating mechanism, then examine the cut-out and note the length of the spark, which should be about $\frac{3}{16}''$ long at full load.

If all these adjustments are right, make the test described above for a short circuit in the fields.

RING ARMATURES.

All K, M, LD and MD machines are now made with ring armatures.

A recent improvement in the construction of these armatures consists in the removal of all insulation from the cores and the addition of more insulation to the separate coils. The cores are divided into three sections with ventilating spaces between.

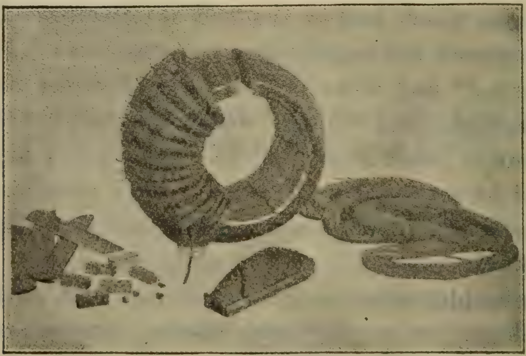


Fig. 123. Armature core and winding.

By removing the insulation from the cores these new coils may be applied to any of the older armatures now in use.

In case it becomes necessary to remove a faulty coil, the following directions should be carefully followed: —

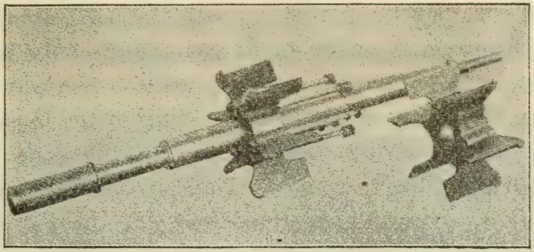


Fig. 124. Armature spider and shaft.

DIRECTIONS FOR PLACING COILS IN RING ARMATURE WITH INSULATED CORE.

After the armature has been taken out of the machine, remove the brass binding wire by cutting the bands, carefully covering all the exposed parts of the armature with a cloth, so as to prevent filings from lodging on the winding. Remove the cord and the tape from the joints of the lead wires and cross connections, at each end of the armature. Take out the lead wires and remove the wooden disks from the shaft. These disks are held in place by a set-screw, passing through a brass piece let into the disk, and resting on the shaft. Unsolder the joints on the coils that are to be removed. Take out the bolts holding the two gun-metal spiders together, and with a long steel pin or drift, drive out the key, which fastens the loose spider to the shaft. The spider next to the pulley is securely fastened to the shaft by a steel pin drawn tightly into a reamed hole, passing through

both spider and shaft. By driving on the commutator end of the shaft with a hard-wood block and mallet, or lead hammer the shaft with the fixed spider may be removed, and the remaining loose spider can then be driven out with the hard-wood block and mallet. Before removing the shaft and spiders note the position of the wedge in the armature core, its position is always indicated by the letter *W* plainly stamped on the hub of the loose spider.

Remove the wood spacing blocks, slip the coils around on the core until the imperfect coils are over the wedge, then spread these coils apart so as to expose the wedge and cut away the insulation on the core for a space of $3\frac{1}{2}''$ on top and bottom over the space containing the wedge; the wedge may then be driven towards the center of the core, taking care that it does not drop on the coils opposite and injure them. The faulty coils may now be removed, new ones be inserted and the wedge be replaced and very carefully reinsulated. This insulation is put on, beginning with the layer next to iron core, as follows: —

(1) 1 layer of paper,
(2) 1 layer of mica,
(3) 1 layer of sheeting,
(4) 1 layer of tape,
(5) 1 layer of mica,
(6) 1 layer of canvas,
(7) 1 layer of tape,
(8) 1 layer of paper.

Slip the coils around to their proper places so that they will be in correct position with regard to the arms of the spiders.

The loose spider may now be put in place, and afterwards the fixed spider and shaft, the bolts being inserted and the nuts tightened up. Replace the key in the loose spider, put on the wooden disks and carefully solder and tape all the joints of lead wires and cross connections. Replace the spacing blocks in their proper positions, solder and tape the connections, and the armature is ready to be bound.

The binding wire used is No. 11, hard brass. The arrangement of the binding wire is clearly shown in the original bands of the armature and should be carefully noted before they are removed. The same brass clips may be used again, provided due care is taken in bending up the ends, when the old band is taken off.

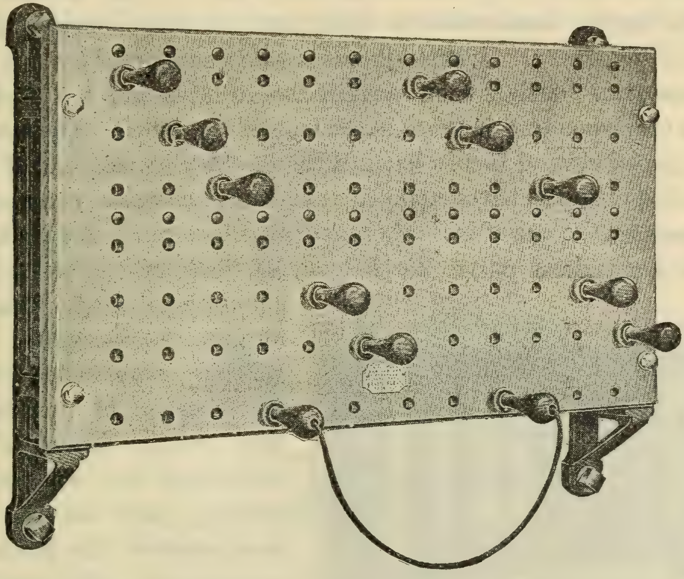


Fig. 125. Standard plug switchboard for 6 circuits.

SWITCHBOARDS.

The standard arc lighting switchboard consists of a marble panel, to the back of which the conductors are attached. When very large boards are built they are made by combining several panels. Switchboards of any capacity can be constructed without

difficulty. The general arrangement of conductors is the same for all sizes. See Figs. 125 and 126.

Each panel is drilled with counter-sunk holes arranged in rows, and in each hole, a brass bushing is fitted. All the bushings of the same horizontal row on the right of the center of the panel are electrically connected, except those of the bottom row, and a similar connection is made between the bushings on the left of the center. A heavy brass strap is supported by the back of the panel behind each vertical row of holes and has bushings in it corresponding to those in the face of the panel. These straps are placed several inches back of the marble, but any one of them can be put in electrical connection with any horizontal conductor it crosses by the use of suitable brass plugs inserted in the bushings.

In a standard panel the number of horizontal rows of holes equals one more than the number of generators. The vertical rows are always twice the number of generators. The positive leads of the generators are attached to binding posts on the left-hand ends of the horizontal conductors. The negative leads are connected to the corresponding binding posts at the right-hand end of the board.

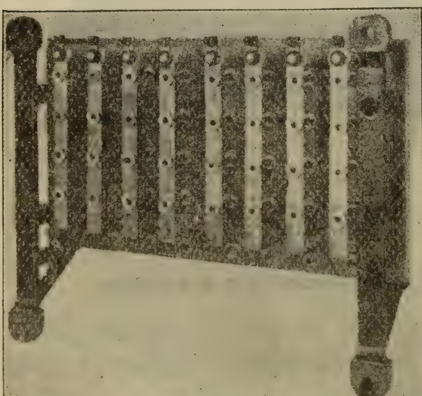


Fig. 126. Back of switchboard.

The positive line wires are connected to the vertical straps on the left, and the negative wires to similar straps on the right of the center of the panel,

If a switchboard plug be inserted in any of the holes of the

board, it puts the corresponding generator lead and line wire in electrical connection, but as the positive line wires are back of the positive generator leads only, it is not possible to reverse the connection of the line and generator accidentally, though any other combinations of lines and generators can be made readily and quickly.

The holes of the lower horizontal row have bushings connected

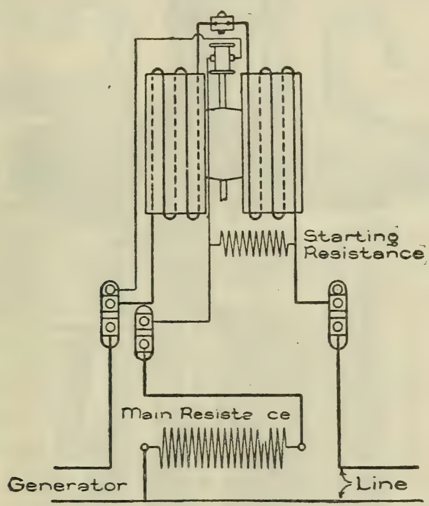


Fig. 127. Meter for station use. Connections for Watt-meters for series arc circuits.

with the vertical straps only. Plugs connected in pairs by flexible cable and inserted in the holes put the corresponding vertical straps in connection as needed, and normally independent lines may be connected when one generator is required to supply several circuits.

Lines and generator leads may be transferred, while running,

by the use of these cables, without shutting down the machines or extinguishing lamps.

WATT-METERS.

Watt-meters are now built to measure the power supplied on series arc circuits. These watt-meters are similar in principle

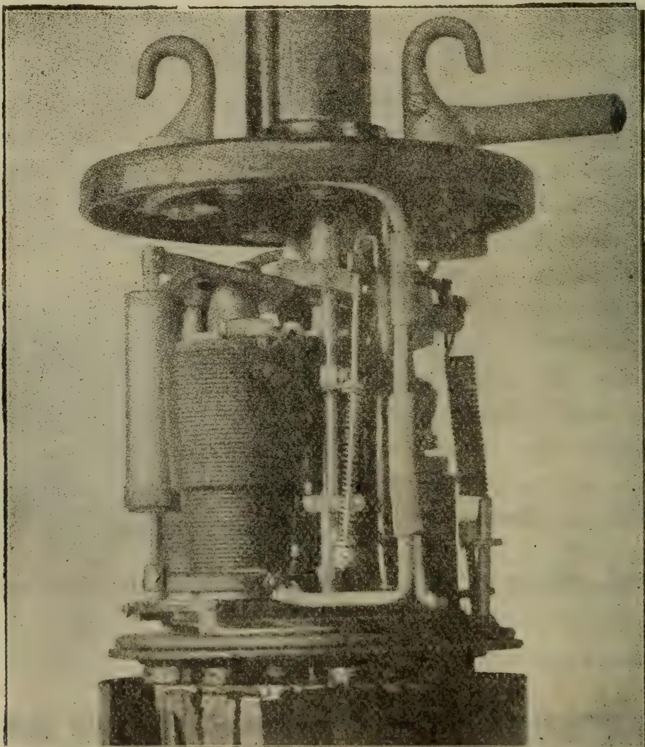


Fig. 128. Interior of M arc lamp.

to those used on incandescent lighting systems, and, being extremely accurate, are equally effective in preventing waste of

current. The watt-meters supplied to customers are made in 4 lamp or 8 lamp capacities. An excess of voltage equivalent to two lamps over the rated load causes the meter to automatically cut out, both lamps and meters being short-circuited. This prevents the interruption of the series circuit in case of any local trouble with lamps or line inside the meter circuit. Station watt-meters are arranged to measure the total output of a generator, and are made with capacities for 35, 50, 65, 80, 125 or 150 lamps. See Fig. 127 for Watt-meter connections.

INSTRUCTIONS FOR THE INSTALLATION AND CARE OF ARC LAMPS.

The lamps should be hung from the hanger boards provided with each lamp, or from suitable supports of wire or chain.

As the hooks on the lamp form also its terminals, they should be insulated, where a hanger board is not used, from the chains or wires used to support the lamp.

To make the upper carbon positive the wire from the positive terminal of the machine should be fastened into the binding post-hook, on the switch side of the *D* lamp, and on the opposite side in the *M* and *K* lamps. When the lamps are hung where they are exposed to the weather, they should be covered with a metal hood, to prevent injury from rain or snow. In such cases care should be taken that the circuit wires do not form a contact on the metal hood, and short-circuit the lamp. Before the lamps are hung up they should be carefully examined to see that the joints are free to move, and that all connections are perfect.

No lamp should be allowed to remain in circuit with the covers removed and mechanism exposed. Such practice is dangerous.

STARTING THE LAMPS.

When the lamps are all in position and ready for operation the machine may be started, and when the armature has reached

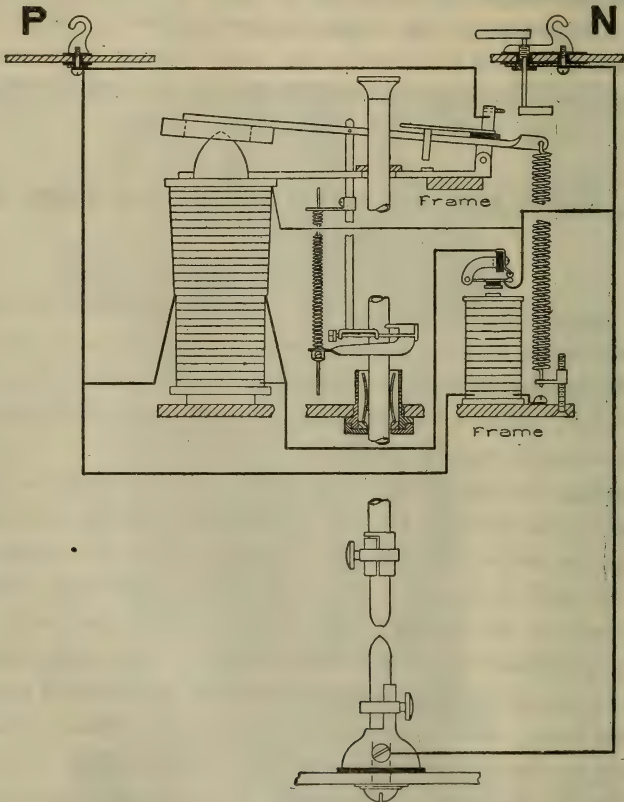


Fig. 129. Connections for M and K arc lamps.

its proper speed, the short-circuiting switch on the frame should be opened.

This method allows the generator to take up its load gradually, and is a very important point in the handling of the machine, particularly when series-incandescent lamps are in the circuit.

The generator should be driven at its proper speed, as designated by the maker. The regulator lever will first rise and then oscillate slowly up and down for short distances, as the regulator is cut in and out by the controller magnet. If the movements are too great, the lights will vary in intensity — first up, then down. This condition will result from a weakness of the regulator dashpot. The regulator lever should always be a short distance away from the stop — say from $\frac{1}{8}''$ to $\frac{1}{2}''$ or more, according to conditions — and should always vibrate up and down in the manner stated. Should the lever of the regulator remain down, it shows that the speed of the machine is not sufficient to supply the circuit, or that the machine is overloaded with lights.

The controller magnet should be constantly opening and closing its contacts. This movement is very slight. The arc of the 2000 c. p. lamps should be $\frac{3}{32}''$ to $\frac{1}{8}''$ in length and the 1200 c. p. lamps should have an arc $\frac{1}{32}''$ to $\frac{1}{16}''$ in length. If the carbons are of good quality, the arc should not flame or hiss.

INSTRUCTIONS FOR REPAIRING, TESTING AND ADJUSTING ARC LIGHTS.

It frequently becomes necessary, after the lamps have been in use for a considerable length of time, to repair and readjust them.

After cleaning and repairing, the lamp should be tested and readjusted. Experience shows that whenever even one new part has been put into a lamp or generator, trouble may result if tests and readjustments are not made before putting the apparatus into regular service.

In order to properly test the lamps that have been repaired, select some part of the engine room where the lamps can be hung

up and burned without being subjected to drafts of air; otherwise, they may hiss and act badly, no matter how carefully the adjustments may be made.

When the lamps have been hung up and attached to the hanger boards, or some similar arrangement for connecting to the circuit in the usual manner, the carbon rods should be cleaned thoroughly with cotton waste. If any sticky or dirty spots appear, which cannot be readily removed with waste, use a piece of well-worn crocus cloth, always being careful to use a piece of clean waste before pushing the rod up into the lamp. Under no circumstances whatever should the rods be pushed up into the lamps in a dirty condition; they should always be cleaned in the manner described.

The tension of the clamp which holds the rod is adjusted by raising or lowering the arm at the top of the guide rod. If the tension is too great, the rod and clutch will wear badly and the feeding will be uneven, causing unsteadiness in the lights. Too light tension will not allow the clutch to hold up the rod and any sudden jar to the lamp will cause the rod to fall and the light to go out.

The double carbon or *M* lamp should have the tension of the second carbon rod a trifle lighter than the first one.

When adjusting the tension, be sure to keep the guide rod perpendicular and in perfect line with the carbon rod; it should be free to move up and down without sticking.

The tension of the clutch in the *D* lamp should be the same as that of the *K* lamp. It is adjusted by tightening or loosening the small coil spring from the arm of the clutch to the bottom of the clamp stop.

To adjust the feeding point in the *K* lamp, press down the main armature as far as it will go, then push up the rod about one-half its length, let go the armature and then press it down slowly, and note the distance of the bottom side of the armature

above the base of the curved part of the pole. When the rod just feeds, this distance should be $\frac{1}{4}''$. If it is not, raise or lower the small stop which slides on the guide rod passing through the arm of the clutch, until the carbon rod will feed when the armature is $\frac{1}{4}''$ from rocker frame at the base of the pole.

To adjust the feeding point of the *M* lamp, adjust the first rod as in the *K* lamp. Then let the first rod down till the cap at the top rests on the transfer lever. The second rod should feed with the armature at a point $\frac{1}{16}''$ higher than it was while feeding the first rod, that is, it should be $\frac{5}{16}''$ from rocker frame at base of pole.

The feeding point of the *D* lamp is adjusted by sliding the clamp stop up or down, so that the rod will feed, when the relative distances of the armature of the lifting magnet and the armature of the shunt magnet from rocker frame are in the ratio of 1 to 2. There should be a slight lateral play in the rocker, between the lugs of the rocker frame.

Make a careful examination of all joints, screws, wires and other parts of the lamps. The armatures of all the magnets should be central with cores, and come down squarely and evenly. There should be a separation of $\frac{1}{32}''$ between the silver contact points, when the armature of the starting magnet is down. This contact should be perfect when the armature is up. The arm for adjusting the tension should not touch the wire or frame of the lamp, when at the highest point. There should be a space of $\frac{3}{32}''$ or $\frac{1}{8}''$ between the body of the clutch and the arm of the clutch, to allow for wear on the bearing surfaces.

Always trim the lamps with carbons of proper length to cut out automatically, that is, have twice as much carbon projecting from the top as from the bottom holder. Always allow a space of $\frac{1}{4}''$ when the lamp is trimmed, from the round head screw in the rod, near the carbon holder, to the edge of upper bushing, so that there will be sufficient space to start the arc Be careful to

get the carbons as accurately centered as possible. They will generally come right after one or two trials.

The arcs of the 1200 candle-power lamps should be adjusted to $\frac{3}{64}''$, with full length of carbon. Arcs of 2000 candle-power lamps should be adjusted from $\frac{1}{16}''$ to $\frac{3}{32}''$ when good carbons are used. Lamps should always maintain a fairly even arc. The length of the arc will slightly increase as the carbons burn away, but they should not hiss, flame, or overfeed at any time. If the switch is thrown and the lamp cut off, and then turned on quickly, the upper carbon should "pick up" promptly with a normal arc, not hiss over a few seconds, and then burn as quietly as before.

When the upper carbon rod is drawn up by the hand, the lamp should cut out promptly and not "flash" the generator. In the case the arc is very long or causes flashing, look at the contacts and see that they are clean and make a good square contact. Also examine the centering of the armature. The cause of the trouble will usually be found in one of these places.

The action of a lamp that feeds badly may often be confounded with that of a badly flaming carbon. The distinction can readily be made after a short observation. The arc of a lamp that feeds badly will gradually grow long until it flames, the clutch will let go suddenly, the upper carbon will fall until it touches the lower carbon, and then pick up. A bad carbon may burn nicely and feed evenly, until a bad spot in the carbon is reached, when the arc will suddenly become long and flame and smoke, due to impurities in the carbon. Instead of dropping as in the former case, the upper carbon will feed to its correct position, without touching the lower carbon.

After the lamp has been tested and burns satisfactorily in the station, tighten up the adjusting screws, and if necessary, put a small amount of thick shellac on the bottom of the guide rod. This will prevent the stop from falling, in case the screw which holds it becomes loose or broken. The lamps are now ready to

be placed on the circuit, but if it is necessary to store them, they should be put into some part of the building or engine room where they will not become covered with dust before they are taken out. If they become dusty, use a small hand bellows to blow away the dust which may have collected on the working parts of the lamps, before placing them on the circuit.

SUMMARY.

The following summary of the foregoing instructions may be useful for the guidance of men in charge of dynamos: —

1. In operating an arc system, attend strictly to all the points herein given.

2. Be sure that the speed of the dynamo is right and that the belt has its proper tension.

3. See that the regulator always works properly, and has sufficient "surplus" or space between its armature and the stop.

4. Be careful that all connections of wires are well made.

5. Do not allow the circuit to become uninsulated at any point.

6. Keep every part of the machine and lamps scrupulously clean.

7. Keep all the insulations free from metallic dust or gritty substances, by a careful cleaning once a day.

8. Keep the bearings of the machine well supplied with the best quality of mineral oil.

9. Do not use water or ice on a bearing in case of heating, as the water is liable to get into the armature and injure the insulation.

10. Lubricate the commutator of the *C* and *E* machines by touching the surface occasionally with an oiled cloth.

11. The commutator on the machine is set carefully before leaving the factory in the best position for proper working, and its position marked by chisel marks on the commutator and shaft.

If the commutator is ever removed from the machine, it must be put back in exactly the same position on the shaft, and the red, white and blue leads must be put into the posts marked 1, 2 and 3 respectively. If wrongly placed, the machine will either not generate, or will act very badly.

12. When the commutator segments become badly worn, they may be turned down in a lathe, either by removing the commutator entirely from the shaft of the machine and putting it upon an arbor, or by removing the segments separately and screwing them to a jig, which may then be put into the lathe. The use of the jig is especially recommended for turning down the segments as the adjustment of the commutator is less liable to be changed than when the arbor is used.

13. The durability of the commutator segments will depend on the care exercised in the running of the machine.

14. The brushes must be set carefully by the "gauge for brushes," in the manner explained before.

15. The spark on the tips of the brushes will vary with the set and wear of the brushes. It should be from $\frac{1}{8}''$ to $\frac{1}{4}''$ long, and only on the forward brushes.

16. The carbon rods in every lamp should be carefully cleaned daily.

17. The carbons should be in perfect alignment and firmly clamped in the holders.

18. If a lamp burns badly and with a bluish flame, or continually hisses, it is probably due to poor carbons, which should be removed and better ones substituted.

19. The lamps rarely burn as well when first started as afterwards. This is principally due to the fact that the carbons require a little time to burn to the proper shape.

20. The automatic regulator prevents the machine from generating more than the amount of current required, so that the lamps may be thrown on or off the circuit at pleasure.

21. Do not tamper with adjustments made in the factory.

22. Do not imagine that every time a lamp hisses or flames a little it is out of adjustment. As a rule, bad working is due to stickiness of the moving parts, or to poor carbons. The lamps once properly adjusted and operated with good carbons, should not get out of adjustment, and should be let alone in that respect.

23. If the machine works badly, it should be tested with a magneto for grounds of connection between the circuit and the frame of the machine. The circuit should also be daily tested, and any faults found should be immediately remedied, as otherwise they will inevitably cause trouble.

24. All construction and repair work should be done in strict accordance with the rules herein laid down.

TABLE OF MAGNETIZING FORCE IN AMPERE TURNS REQUIRED PER INCH OF LENGTH OF MAGNETIC CIRCUIT.

Magnetic Density per square inch in Gausses.	MAGNETIZING FORCE IN AMPERE TURNS.			
	Air.	Cast Iron.	Steel.	Wrought Iron.
5,000	1,567	3.80	2.85	1.50
10,000	3,134	5.35	4.25	2.40
15,000	4,701	6.80	5.35	3.20
20,000	6,268	8.00	6.30	3.90
25,000	7,835	10.30	7.50	4.60
30,000	9,402	16.20	8.80	5.30
35,000	10,969	28.70	10.20	5.90
40,000	12,536	49.00	11.70	6.50
45,000	14,103	80.00	13.40	7.10
50,000	15,670	160.00	15.40	8.20
55,000	17,237	240.00	17.80	9.50
60,000	18,804	350.00	20.70	11.00
65,000	20,371	490.00	24.10	13.50
70,000	21,938	650.00	28.00	17.00
75,000	23,505	....	34.00	21.80
80,000	25,072	....	42.00	27.50

CHAPTER Xa.

Incandescent Wiring Table.

Table on two following pages is arranged to enable wiremen to select the right sizes of wire for service connections and inside work. The figures at the top indicate distance in feet to center of distribution, in reality half the length of the circuit; the four columns at the left showing the number of 16-candle power lamps at various voltages; the other figures showing the sizes of wire, Brown & Sharpe gauge, to be used for distributing the number of lamps stated at the distances indicated and with the loss of 1 volt.

For example: To distribute 30 lamps of 110 volts at a distance of 80 feet with a loss of 1 volt. In column of 110-volt lamps find the number 30, then follow the same line of figures to the right until the column headed 80 is reached, and it appears that No. 6 wire must be used.

The same table may be used for other losses than 1 volt by dividing the given number of lamps by the number of volts to be lost, then with this product proceed as before in the table.

For example: To distribute 30 lamps of 110 volts at a distance of 80 feet with a loss of 2 volts, divide 30 by 2 which gives 15, then find 15 in the column headed 110 volts and follow the same line of figures to the right until column headed 80 is reached, and it is found that No. 8 wire must be used.

No wire smaller than No. 14 is shown in the table as the National Board of Fire Underwriters prohibits the use of a smaller size. Odd sizes smaller than No. 5 are not commercial and are therefore omitted.

Incandescent Wiring Table.

Sixteen Candle Power Lamps. Loss, One Volt.

TABLE No. 1. Sizes of Wire are by B. & S. Gauge.

52 Volt $3\frac{1}{2}$ Watt Lamps	110 Volt $3\frac{1}{2}$ Watt Lamps	220 Volt 4 Watt Lamps	550 Volt 4 Watt Lamps	Distance in feet to center of Distribution.					
				20′	25′	30′	35′	40′	45′
1	2	3	9	14	14	14	14	14	14
2	4	7	18	14	14	14	14	14	14
3	6	11	28	14	14	14	14	14	14
4	8	15	37	14	14	14	14	14	14
5	10	18	46	14	14	14	14	12	12
6	12	22	56	14	14	14	12	12	12
7	15	26	65	14	14	12	12	12	10
8	17	30	74	14	12	12	12	10	10
9	19	33	83	14	12	12	10	10	10
10	21	37	93	12	12	12	10	10	10
12	25	44	111	12	10	10	10	8	8
14	30	52	130	12	10	10	8	8	8
16	34	59	148	10	10	8	8	8	8
18	38	66	167	10	8	8	8	8	6
20	42	74	185	10	8	8	8	6	6
25	53	92	232	8	8	6	6	6	6
30	63	111	278	8	6	6	6	5	5
35	74	130	324	6	6	6	5	5	4
40	85	148	371	6	6	6	5	4	4
45	95	166	428	5	5	5	4	4	3
50	106	185	464	5	5	4	4	3	3
55	116	203	510	4	4	4	3	3	2
60	127	222	557	4	4	4	3	2	2
65	138	240	603	3	3	3	3	2	2
70	148	260	650	3	3	3	2	2	1
75	159	277	696	2	2	2	2	1	1
80	170	296	742	2	2	2	2	1	1
90	191	333	835	1	1	1	1	1	0
100	212	370	928	1	1	1	1	0	0

Incandescent Wiring Table.

Sixteen Candle Power Lamps, Loss, One Volt.

TABLE No. 1a. Sizes of Wire are by B. & S. Gauge.

DISTANCE IN FEET TO CENTER OF DISTRIBUTION.

50′	60′	70′	80′	90′	100′	120′	140′	160′	180′	200′
14	14	14	14	14	14	14	14	14	14	12
14	14	14	14	14	12	12	12	10	10	10
14	14	12	12	12	10	10	10	8	8	8
12	12	12	10	10	10	8	8	8	8	6
12	10	10	10	10	8	8	8	6	6	6
10	10	10	8	8	8	8	6	6	6	6
10	10	8	8	8	8	6	6	6	5	5
10	8	8	8	8	6	6	6	5	5	4
10	8	8	8	6	6	6	5	5	4	4
8	8	8	6	6	6	5	5	4	4	3
8	8	6	6	6	5	5	4	3	3	2
8	6	6	6	5	5	4	3	3	2	2
6	6	6	5	5	4	3	3	2	2	1
6	6	5	5	4	4	3	2	2	1	1
6	5	5	4	4	3	2	2	1	1	0
5	5	4	3	3	2	1	1	0	0	00
5	4	3	2	2	1	1	0	0	00	00
4	3	2	2	1	1	0	00	00	000	000
3	2	2	1	1	0	00	00	000	000	0000
3	2	1	1	0	0	00	000	000	0000	0000
2	1	1	0	0	00	000	000	0000	0000	
2	1	0	0	00	00	000	0000	0000		
1	1	0	00	00	000	000	0000	0000		
1	0	0	00	00	000	0000	0000			
1	0	00	00	000	000	0000				
0	0	00	000	000	0000	0000				
0	00	00	000	000	0000					
0	00	000	000	0000	0000					
00	000	000	0000	0000						

Feet x 2 x 10.70.

TABLE NO. 2.

Feet to end of Circuit.	Ft. x2x10.70.	Feet to end of Circuit.	Ft. x2x10.70.	Feet to end of Circuit.	Ft. x2x10.70.
5	107	185	3,959	365	7,811
10	214	190	4,066	370	7,918
15	321	195	4,173	375	8,025
20	428	200	4,280	380	8,132
25	535	205	4,387	385	8,239
30	642	210	4,494	390	8,346
35	749	215	4,601	395	8,453
40	856	220	4,708	400	8,560
45	963	225	4,815	405	8,667
50	1,070	230	4,922	410	8,774
55	1,177	235	5,029	415	8,881
60	1,284	240	5,136	420	8,988
65	1,391	245	5,243	425	9,095
70	1,498	250	5,350	430	9,202
75	1,605	255	5,457	435	9,309
80	1,712	260	5,564	440	9,416
85	1,819	265	5,671	445	9,523
90	1,926	270	5,778	450	9,630
95	2,033	275	5,885	455	9,737
100	2,140	280	5,992	460	9,844
105	2,247	285	6,099	465	9,951
110	2,354	290	6,206	470	10,058
115	2,461	295	6,313	475	10,165
120	2,568	300	6,420	480	10,272
125	2,675	305	6,527	485	10,379
130	2,782	310	6,634	490	10,486
135	2,889	315	6,741	495	10,593
140	2,996	320	6,848	500	10,700
145	3,103	325	6,955	510	10,914
150	3,210	330	7,062	520	11,128
155	3,317	335	7,169	530	11,342
160	3,424	340	7,276	540	11,556
165	3,531	345	7,383	550	11,770
170	3,638	350	7,490	560	11,984
175	3,745	355	7,597	570	12,198
180	3,852	360	7,704	580	12,412

Feet x 2 x 10.70.

TABLE NO. 2.

Feet to end of Circuit.	Ft. x2x10.70.	Feet to end of Circuit.	Ft. x2x10.70.	Feet to end of Circuit.	Ft. x2x10.70.
590	12,626	970	20,758	1,350	28,890
600	12,840	980	20,972	1,360	29,104
610	13,054	900	21,186	1,370	29,318
620	13,268	1,000	21,400	1,380	29,532
630	13,482	1,010	21,614	1,390	29,746
640	13,696	1,020	21,828	1,400	29,960
650	13,910	1,030	22,042	1,410	30,174
660	14,124	1,040	22,256	1,420	30,388
670	14,338	1,050	22,470	1,430	30,602
680	14,552	1,060	22,684	1,440	30,816
690	14,766	1,070	22,898	1,450	31,030
700	14,980	1,080	23,112	1,460	31,244
710	15,194	1,090	23,326	1,470	31,458
720	15,408	1,100	23,540	1,480	31,672
730	15,622	1,110	23,754	1,490	31,886
740	15,836	1,120	23,968	1,500	32,100
750	16,050	1,130	24,182	1,510	32,314
760	16,264	1,140	24,396	1,520	32,528
770	16,478	1,150	24,610	1,530	32,742
780	16,692	1,160	24,824	1,540	32,956
790	16,906	1,170	25,038	1,550	33,170
800	17,120	1,180	25,252	1,560	33,384
810	17,334	1,190	25,466	1,570	33,598
820	17,548	1,200	25,680	1,580	33,812
830	17 762	1,210	25,894	1,590	34,026
840	17,976	1,220	26,108	1,600	34,240
850	18 190	1,230	26,322	1,610	34,454
860	18,404	1,240	26,536	1,620	34,668
870	18,618	1,250	26,750	1,630	34,882
880	18,832	1,260	26,964	1,640	35,096
890	19,046	1,270	27,178	1,650	35,310
900	19,260	1,280	27,392	1,660	35,524
910	19,474	1,290	27,606	1,670	35,738
920	19,688	1,300	27,820	1,680	35,952
930	19,902	1,310	28,034	1,690	36,166
940	20,116	1,320	28,248	1,700	36,380
950	20,330	1,330	28,462	1,710	36,594
960	20,544	1,340	28,676	1,720	36,808

Feet x 2 x 10.70.

TABLE NO. 2.

Feet to end of Circuit.	Ft. x2x10.70.	Feet to end of Circuit.	Ft. x2x10.70.	Feet to end of Circuit.	Ft. x2x10.70.
1,730	37,022	2,450	52,430	4,250	90,950
1,740	37,236	2,500	53,500	4,300	92,020
1,750	37,450	2,550	54,570	4.350	93,090
1,760	37,664	2,600	55,640	4,400	94,160
1,770	37,878	2,650	56,710	4,450	95,230
1,780	38,092	2,700	57,780	4,500	96,300
1,790	38,306	2,750	58,850	4,550	97,370
1,800	38,520	2,800	59,920	4,600	98,440
1,810	38,734	2,850	60,990	4,650	99,510
1,820	38,948	2,900	62,060	4,700	100,580
1,830	39,162	2,950	63,130	4,750	101,650
1,840	39,376	3,000	64,200	4,800	102,720
1,850	39.590	3,050	65,270	4,850	103,790
1,860	39,804	3,100	66,340	4,900	104,860
1,870	40,018	3,150	67,410	4,950	105,930
1,880	40,232	3,200	68,480	5,000	107,000
1,890	40,446	3,250	69,550	5,050	108,070
1,900	40,660	3,300	70,620	5,100	109,140
1,910	40,874	3,350	71,690	5,150	110,210
1,920	41.088	3,400	72,760	5,200	111,280
1,930	41,302	3,450	73,830	5,250	112,350
1,940	41,516	3,500	74,900	5,300	113,420
1,950	41,730	3,550	75,970	5,350	114,400
1,960	41,944	3,600	77,040	5,400	115,560
1,970	42,158	3,650	78,110	5,450	116,630
1,980	42,372	3,700	79,180	5,500	117,700
1,990	42,586	3,750	80,250	5,550	118,770
2,000	42.800	3,800	81,320	5,600	119,840
2,050	43,870	3,850	82,390	5,650	120,910
2,100	44,940	3,900	83,460	5,700	121,980
2,150	46,010	3,950	84,530	5.750	123,050
2,200	47,080	4,000	85,600	5,800	124,120
2,250	48,150	4,050	86,670	5,850	125,190
2,300	49,220	4,100	87,740	5,900	126,260
2,350	50,290	4,150	88,810	5,950	127,330
2,400	51,360	4,200	89,880	6,000	128,400

Feet x 2 x 10.70.

TABLE NO. 2.

Miles.	Ft. x 2 x 10.70	Miles.	Ft. x 2 x 10.70	Miles.	Ft. x 2 x 10.70
½	564,960	4	451,968	7½	847,440
1	112,992	4½	508,464	8	903,936
1½	169,488	5	564,960	8½	960,432
2	225,984	5½	621,456	9	1,016,928
2½	282,480	6	677,952	9½	1,073,424
3	338,976	6½	734,448	10	1,129,920
3½	395,472	7	790,944		

$$\text{(A)}\quad \frac{\text{Feet x 2 x 10.7 x Amperes}}{\text{Volts lost}} = \text{Circular mils.}$$

$$\text{(B)}\quad \frac{\text{Feet x 2 x 10.7 x Amperes}}{\text{Circular mils.}} = \text{Volts lost.}$$

$$\text{(C)}\quad \frac{\text{Circular mils x volts lost}}{\text{Feet x 2 x 10.7}} = \text{Amperes.}$$

In calculating the sizes of wire as shown in the Incandescent Wiring Table a formula (A) has been used in which there is a constant 10.7, the number of circular mils in a copper wire which would have a resistance of one ohm for one foot of length. One ampere through one ohm resistance loses one volt. To determine the size of wire necessary for carrying a given current a given distance in feet, multiply the number of feet by 2 to obtain the actual length of circuit, multiply this product by the constant 10.7 and it will give the circular mils necessary for one ohm resistance, multiply this by the amperes and it gives the circular mils necessary for the loss of one volt. Divide this last result by the volts lost and it gives the circular mils necessary. Hence the formula "A."

By simply transposing the terms we obtain formula "B," which can be used to determine the volts lost in a given length of wire of certain size carrying a certain number of amperes.

Again, by another change in the terms, we obtain formula "C," which shows the number of amperes which a wire of given size and length will carry at a given number of volts lost.

Table No. 2 has been arranged for the purpose of saving time in the use of these formulas. It shows the result of Feet x 2 x 10.7 for various distances over which it may be desired to transmit current.

A few examples will assist in showing the use of the formulas and tables.

Suppose we wish to distribute 300 16 c. p. 3.5 watt lamps of 110 volts at a distance of 490 feet with a loss of 10 per cent.

Using formula A,

490 feet x 2 x 10.7 (find it in table No. 2) = 10486.

300 lamps of 110 volts = 152.7 amperes.

(See table No. 3 for amperes per lamp, and multiply by 300.)

10 per cent loss on 110 volt system = 12.22 volts. (See table No. 4.)

10486 x 152.7 amperes = 1601212 circ. mils. ÷ 12.22 volts lost = 131032 circ. mils.

In our table it shows the size of wire for this number of circ. mils. to be 00.

To check this and determine exactly the volts lost in this circuit by using No. 00 wire, use formula B, as follows:

10,486 x 152.7 amperes = 1601212 ÷ 133079 circ. mils. = 12.03 volts lost.

Suppose it is desired to distribute 1,000 lamps at a distance of 1950 feet by 3-wire system, viz., 220 volts, with a loss of 10 per cent.

Using formula A,

1950 feet x 2 x 10.7 (see table) = 41730.

1000 lamps on 220 volt system = 291 amperes.

(See table No. 5 for amperes per lamp, and multiply by 1000.)

10 per cent on 220 volt system = 24.44 volts lost. (See table No. 4.)

41730 x 291 amperes = 12143430 ÷ 24.44 volts lost = 496867 circ. mils.

500000 circ. mils, the nearest commercial size, should be used.

Check this as before by formula B.

41730 x 291 amperes = 12143430 ÷ 500000 circ. mils = 24.29 volts lost.

Suppose we wish to deliver 100 h. p. to a 500 volt motor, at a distance of 4850 feet with 10 per cent loss:

Again using formula A,

4850 feet x 2 x 10.7 = 103790.

100 h. p. at 500 volts = 160 amperes. (See table No. 3.)

10 per cent loss on 500 volts system = 55.5 volts. (See table No. 4.)

103790 x 160 amperes = 16606400 ÷ 55.5 volts = 299215 circ. mils.

300000 circ. mils cable should be used.

Check this as before by formula B.

103790 x 160 amperes = 16606400 ÷ 300000 circ. mils = 55.35 volts lost.

To ascertain how many amperes could be carried to a distance of 4850 feet with 500 volts with 10 per cent loss, use formula C:

4850 feet x 2 x 10.7 = 103790.

10 per cent loss on 500 volts system = 55.5 volts.

300000 circ. mils x 55.5 volts lost ÷ 103790 = 160.42 amperes, which as will appear by reference to table No. 3, will permit the use of 100 h. p. motor.

Amperes per Motor.

TABLE No. 3.

H. P.	Per Cent Efficiency	Watts.	VOLTS.		
			110	115	120
¾	65	860	7.82	7.48	7.17
1	65	1148	10.4	9.98	9.57
2	65	2295	20.8	20.0	19.1
2½	75	2487	22.6	21.6	20.7
3½	75	3480	31.6	30.3	29.0
5	80	4662	42.4	40.5	38.8
7½	80	6994	63.6	60.8	58.3
10	85	8776	79.8	76.3	73.1
15	85	13165	120.	114.	110.
20	90	16578	151.	144.	138.
25	90	20722	188.	180.	173.
30	90	24867	226.	216.	207.
40	90	33155	301.	288.	276.
50	90	41444	377.	360.	345.
70	90	58022	528.	505.	484.
90	90	74600	678.	649.	622.
100	93	80215	729.	697.	668.
125	93	100269	912.	872.	836.
150	93	120323	1094.	1046.	1003.

The above table is arranged to show the amperes per motor at different voltages for several sizes of motors at efficiencies obtained in ordinary practice.

Amperes per Motor.

TABLE NO. 3.

VOLTS.					
125	220	250	500	525	550
6.88	3.91	3.44	1.72	1.64	1.56
9.18	5.22	4.59	2.30	2.19	2.09
18.4	10.4	9.18	4.59	4.37	4.17
19.9	11.3	9.95	4.97	4.74	4.52
27.8	15.8	13.9	6.96	6.63	6.33
37.3	21.2	18.6	9.32	8.88	8.48
56.0	31.8	28.0	14.0	13.3	12.7
70.2	39.9	35.1	17.6	16.7	16.0
105.	59.8	52.6	26.3	25.1	23.9
133.	75.4	66.3	33.2	31.6	30.1
166.	94.2	82.9	41.4	39.5	37.7
199.	113.	99.4	49.7	47.4	45.2
265.	151.	133.	66.3	63.2	60.3
332.	188.	166.	82.9	79.0	75.4
464.	264.	232.	116.	111.	106.
597.	339.	298.	149.	142.	136.
642.	365.	321.	160.	153.	146.
802.	456.	401.	200.	191.	182.
963.	547.	481.	241.	229.	219.

The above table is arranged to show the amperes per motor at different voltages for several sizes of motors at efficiencies obtained in ordinary practice.

Volts Lost at Different Per Cent Drop.

Voltage at Lamp or Distribution Point, Top Row.

TABLE NO. 4.

VOLTS	52	75	100	110	220	400
½%	.261	.376	.502	.552	1.10	2.01
1%	.525	.757	1.01	1.11	2.22	4.04
1½%	.787	1.14	1.52	1.67	3.35	6.09
2%	1.06	1.53	2.04	2.24	4.48	8.16
2½%	1.33	1.92	2.56	2.82	5.64	10.25
3%	1.61	2.31	3.09	3.40	6.80	12.37
4%	2.16	3.12	4.16	4.58	9.16	16.06
5%	2.73	3.94	5.26	5.78	11.57	21.05
6%	3.31	4.78	6.38	7.02	14.04	25.53
7%	3.91	5.64	7.52	8.27	16.55	30.10
8%	4.52	6.52	8.69	9.56	19.13	34.78
9%	5.14	7.41	9.89	10.87	21.75	39.56
10%	5.77	8.33	11.11	12.22	24.44	44.44
12%	7.09	10.22	13.63	14.99	29.99	54.54
13%	7.76	11.10	14.04	16.43	32.87	59.76
14%	8.46	12.20	16.27	17.90	35.81	65.1
15%	9.17	13.23	17.64	19.41	38.82	70.5
20%	13.	18.75	25.	27.50	55.	100.
25%	17.33	25.	33.33	36.66	73.33	133.

The above table shows the loss in voltage between dynamos and distribution point at different per cents and for various voltages.

Volts Lost at Different Per Cent Drop.

Voltage at Lamp or Distribution Point, Top Row.

TABLE No. 4.

500	600	800	1000	1200	2000
2.51	3 01	4.02	5 02	6.03	10.05
5 05	6.66	8.08	10.10	12.12	20.2
7.61	9.13	12.1	15 2	18.2	30.4
10 2	12.2	16.3	20 4	24 4	40.8
12 8	15 3	20.5	25.6	30.7	51.2
15.4	18.5	24.7	30.9	37.1	61.8
20.8	24.9	33.3	41.6	49 9	83.3
26.3	31.5	42.1	52.6	63.1	105.
31.9	38.2	51.	63.8	76.5	127.
37.6	45.1	60.2	75 2	90.3	150.
43.4	52.1	69.5	86.9	104.	173.
49.4	59.3	79.1	98.9	118.	197.
55 5	66.6	88.8	111.	133.	222.
61.7	74.1	98 8	123.	148.	247.
68.1	81.8	109.	136.	163.	272.
74.7	89.6	119.	149.	179.	298.
81.3	97.6	130.	162.	195.	325.
88.2	105.	141.	176.	211.	352.
125.	150.	200.	250.	300.	400.
166.	200.	266.	333.	400.	666.

By adding the volts given in the table to the voltage at motor or lamp the result shows the voltage necessary at dynamo for voltage required at point of distribution.

Amperes per Lamp.

TABLE NO. 5.

The following table is arranged to show the amperes per lamp for lamps of different candle powers and efficiencies at various voltages. The upper row of figures shows the voltage, the second shows the watts per candle power, or efficiency, and the figures below show the corresponding amperes per lamp for different candle powers.

This table is made in accordance with the best information obtainable from manufacturers on the efficiency of standard lamps in use. Lamps of other efficiencies are on the market, but those shown are standard for good practice at the present time.

															Series Ry. Lamps.		
Watts per C. P.	50 Volts.			52 Volts.			100 Volts.		104 Volts.		110 Volts.		220 V.	500 V.	550 V.	600 V.
	3.1	3.5	4.0	3.1	3.5	4.0	3.1	3.5	3.1	3.5	3.1	3.5	4.0	4.0	4.0	4.0
8 C. P.	.496	.56	.64	.477	.538	.615	.248	.280	.238	.269	.225	.255	.145	.064	.058	.053
10 C. P.	.620	.70	.80	.596	.673	.769	.310	.350	.298	.337	.282	.318	.182	.080	.073	.067
16 C. P.	.992	1.12	1.28	.954	1.077	1.231	.496	.560	.477	.538	.451	.509	.291	.128	.116	.107
20 C. P.	1.240	1.40	1.60	1.192	1.346	1.538	.620	.700	.596	.673	.564	.636	.363	.160	.145	.133
24 C. P.	1.488	1.68	1.92	1.431	1.615	1 846	.744	.840	.715	.808	.676	.764	.436	.192	.175	.160
32 C. P.	1.984	2.24	2.56	1.908	2.154	2.461	.992	1.120	.954	1 077	.902	1 018	.582	.256	.233	.213

Approximate Weight and Measurement of "O. K." Triple Braided Weatherproof Copper Wire.

TABLE NO. 6.

B. & S. Gauge No.	Feet per Pound.	Pounds per 1000 ft.	Pounds Per Mile.
0000	1.30	767	4050
000	1.59	629	3320
00	2.02	495	2610
0	2.45	407	2150
1	3.22	310	1640
2	4.00	250	1320
3	5.03	199	1050
4	6.10	164	865
5	7.43	135	710
6	9.00	111	587
8	13.54	74	390
10	18.85	53	280
12	28.54	35	185
14	40.61	25	130
16	60.00	17	88
18	75 43	13	70

Table Showing Difference Between Wire Gauges in Decimal Parts of an Inch.

TABLE NO. 7.

No. of Wire Gauge.	American or Brown & Sharpe.	Birmingham or Stubs'.	Washburn & Moen Manufacturing Co., Worcester, Mass.	Trenton Iron Co., Trenton, N. J.	New British.	Old English from Brass Mfrs. List.	No. of Wire.
000000	....	....	.46	....	....	....	000000
00000	....	......	.43	.45	....	....	00000
0000	.46	.454	.393	.4	.4	....	0000
000	.40964	.425	.362	.36	.372	....	000
00	.3648	.38	.331	.33	.348	....	00
0	.32495	.34	.307	.305	.324	....	0
1	.2893	.3	.283	.285	.3	....	1
2	.25763	.284	.263	.265	.276	....	2
3	.22942	.259	.244	.245	.252	....	3
4	.20431	.238	.225	.225	.232	....	4
5	.18194	.22	.207	.205	.212	....	5
6	.16202	.203	.192	.19	.192	....	6
7	.14428	.18	.177	.175	.176	....	7
8	.12849	.165	.162	.16	.16	....	8
9	.11443	.148	.148	.145	.144	....	9
10	.10189	.134	.135	.13	.128	....	10
11	.090742	.12	.12	.1175	.116	....	11
12	.080808	.109	.105	.105	.104	....	12
13	.071961	.095	.092	.0925	.092	....	13
14	.064084	.083	.08	.08	.08	.083	14
15	.057068	.072	.072	.07	.072	.072	15
16	.05082	.065	.063	.061	.064	.065	16
17	.045257	.058	.054	.0525	.056	.058	17
18	.040303	.049	.047	.045	.048	.049	18
19	.03589	.042	.041	.039	.04	.04	19
20	.031961	.035	.035	.034	.036	.035	20
21	.028462	.032	.032	.03	.032	.0315	21
22	.025347	.028	.028	.27	.028	.0295	22
23	.022571	.025	.025	.024	.024	.027	23
24	.0201	.022	.023	.0215	.022	.025	24
25	.0179	.02	.02	.019	.02	.023	25
26	.01594	.018	.018	.018	.018	.0205	26
27	.014195	.016	.017	.017	.0164	.01875	27
28	.012641	.014	.016	.016	.0148	.0165	28
29	.011257	.013	.015	.015	.0136	.0155	29
30	.010025	.012	.014	.014	.0124	.01375	30
31	.008928	.01	.0135	.013	.0116	.01225	31
32	.00795	.009	.013	.012	.0108	.01125	32
33	.00708	.008	.011	.011	.01	.01025	33
34	.006304	.007	.01	.01	.0092	.0095	34
35	.005614	.005	.0095	.009	.0084	.009	35
36	.005	.004	.009	.008	.0076	.0075	36
37	.004453	....	.0085	.00725	.0068	.0065	37
38	.003965	....	.008	.0065	.006	.00575	38
39	.003531	....	.0075	.00575	.0052	.005	39
40	.003144	....	.007	.005	.0048	.0045	40

TABLE NO. 8.

Electric Light Conductors.

Brown & Sharpe.	Diameter.		Square of Diameter.	Sectional Area.	Safe Carrying Capacity in Amperes.				Weight.				Length.	
							Chicago Code.		Bare.		Weather-proof insulation.		Bare.	Weather-proof insulation.
No.	Mils.	Millimeters.	Circular Mils.	Square Millimeters.	National Code	New England Code.	Conceal-ed Work	Open Work.	Pounds per 1,000 Feet.	Pounds per Mile.	Pounds per 1,000 Feet.	Pounds per Mile.	Feet per Pound.	Feet per Pound.
0000	460 000	11.684	211600.0	107.219	175	300	218	312	639 33	3359	758	4000	1.6	1.25
000	409.640	10 405	167805.0	85.028	145	245	181	262	507.01	2663	606	3200	2	1.75
00	364.800	9.266	133079.4	67.431	120	215	150	220	402.09	2112	492	2600	2.5	2
0	324.950	8.254	105592.5	53.504	100	190	125	185	319.04	1661	385	2036	3 2	2 50
1	289 300	7 348	83694.2	42.409	95	460	105	156	252.88	1328	308	1628	4	3.25
2	257.630	6.544	66373 0	33.632	70	135	88	131	200.54	1069	250	1310	5	4
3	229 420	5 827	52634.0	26.670	60	115	75	110	159 03	835	201	1060	6 3	5
4	204.310	5 189	41742 0	21.151	50	100	63	92	126.12	662	163	862	8	6
5	181.940	4.621	33102.0	16.773	45	90	53	77	100.01	526	134	706	10	7.50
6	162 020	4.115	26250 5	13.301	35	80	45	65	79.32	417	110	577	12.6	9
7	144.280	3 665	20816 0	10.548	30	67	..		62 90	330	90	475	16	11
8	128.490	3.264	16509.0	8.366	25	60	33	46	49.80	262	74	392	20	13.50
9	114.430	2 907	13094.0	6.635	..	50	.	..	39 56	208	62	328	25	16
10	101.890	2 588	10381 0	5.260	20	40	25	32	31 37	165	52	275	32	19
11	90.742	2.305	8234.0	4.172	..	35	.	..	24 88	131	44	231	40	22.75
12	80.808	2 053	6529.9	3.309	15	30	17	23	19.73	103	36	193	51	27.25
13	71 961	1 828	5178.4	2.618	..	26	.	..	15.65	82	31	162	64	32 50
14	64.084	1.628	4106.8	2.081	10	22	12	16	12.41	65	26	135	81	39
15	57.068	1.450	3256.7	1.650	..	18	..	..	9.84	51.5	21	111	102	47.50
16	50.820	1.291	2582 9	1.309	5	15	..	..	7.81	41	18	96	129	55

THE STEAM ENGINE.

CHAPTER XI.

THE SELECTION OF AN ENGINE.

There are so many conflicting statements in regard to the merits and demerits of the several engines placed in the market that one is often confused in judgment, and scarcely knows how to proceed in the matter of selection,

It is easy to advise that "When you are ready to buy, select the best engine, for in the long run the best is the cheapest." No one would pretend to deny this as a general rule, yet there are circumstances which so materially modify this rule that it would seem to a casual observer to be entirely set aside. There are localities in which the price of fuel is so low that it scarcely warrants the doubling of the price on an engine to save it; and in such localities the owners usually want an engine of the very simplest construction; hence, they almost invariably select an ordinary slide valve engine with a throttling governor. This selection is made for several reasons, among which are low first cost, simple in detail, remoteness from the manufacturer or from repair shops.

For small powers in which it is desirable that the investment be as low as consistent with commercial success, the engine selected should be fitted with a common slide valve; this will in general apply to all engines having cylinders eight inches or less in diameter.

If upon a thorough canvass of the situation, it then be thought advisable to employ an automatic cut-off engine, the next question would probably be whether it shall be fitted with a positive, or some one of the various "drop" movements now in the market.

For the smaller sizes, say 8 to 24 inches diameter of cylinder, it will perhaps be found more desirable to use an automatic slide cut-off, of which there are now several varieties offered through the trade. This style of engine has the advantage of being low-priced, efficient and economical.

Small engines are usually required to run at pretty high speed; there is a very decided advantage in this on the score of economy, as a small engine running at a high speed will be quite as efficient as a large engine running at a slow speed, with the further advantage that the former will not cost in original outlay more than about two-thirds of the latter, while the cost of operating will be no greater per indicated horse-power.

The slide valve is still used to the almost total exclusion of all other kinds in locomotives. It is doubtful whether a better valve for that particular use can be devised. It is simple, efficient, and readily obeys the action of the link when controlled or adjusted by the engineer. For portable engines and the smaller stationary engines it leaves little to be desired in point of simplicity.

One objection to a slide valve is that it cannot readily be made to cut off steam at, say, half-stroke or less, without interfering with the exhaust. In ordinary practice $\frac{5}{8}$ to $\frac{2}{3}$ seems to be where most slide valves cut off as a minimum, perhaps $\frac{3}{4}$ would represent nearer the actual average conditions.

It can easily be shown that this is very wasteful of steam, and consequently not economical in fuel; but as there are cases in which the loss in fuel is fully gained by other advantages, the ordinary slide valve will, in all probability, continue to be used.

High speed engines. — The general tendency seems now to be in the direction of a horizontal engine with a stroke of medium length having a rapid piston speed and a rapid rotation of crank shaft, rather than a longer stroke with a less rate of revolution. This rapid movement of piston and crank shaft permits the use of

small fly-wheels and driving pulleys, and thus very materially reduces the cost of an engine for a given power.

To illustrate this, it may be said that a 16 x 48 inch engine using steam at 80 lbs. pressure and cutting off $\frac{1}{4}$ stroke, running at the rate of 60 revolutions per minute, may be replaced by an engine having a 13 x 24 inch cylinder, running at the rate of 200 revs. per minute, the pressure of steam and point of cutting off remaining the same, both engines being non-condensing, and representing the best examples of their kind. The difference between 60 and 200 revolutions per minute in millwright work is very great, but there is a constantly growing demand for an engine which shall meet such a requirement whenever it shall present itself; by this is not to be understood an engine which shall be used at either speed indiscriminately, but rather a type of engine which shall be economical in fuel, and shall be of a kind by which the rate of revolution may be such as to suit the millwright's work without loss of economy in working, and without excessive outlay for the engine itself in proportion to power developed.

Slow speed engines are designed and built from a standpoint entirely different from that of high speed engines; in the former case the reciprocating parts are made as light as possible, consistent with safety. The fly-wheel is large in diameter and made with a very heavy rim, especially is this the case with automatic cut-off engines of long stroke and slow revolution of crank shaft.

In high speed engines the reciprocating parts are often of great weight, in order to insure the utmost smoothness of running. The piston and cross-head are made of unusual weight that at the beginning of the stroke they may require a large part of the steam pressure to set them in motion; this absorbing of power at the beginning of the stroke is for the purpose of temporarily storing it up in the reciprocating parts that it may be given off at the

later portions of the stroke, by imparting their momentum to the crank; thus at the beginning of the stroke, these reciprocating parts act as a temporary resistance, but once in motion they tend by their inertia to equalize the pressure on the crank pin, and so produce not only smooth running, but a very uniform motion.

Results to be obtained in practice. — The best automatic non-condensing engines furnish an indicated horse-power for about three pounds of good coal, depending somewhat upon the fitness of the engine for the work and the quality of the coal. With a condenser attached, a consumption as low as two pounds has been reported, but this is an exceptional result, $2\frac{1}{2}$ pounds may be quoted as good practice. The larger the engine the better the showing, as compared with smaller engines.

For ordinary slide valve engines, the coal burned per indicated horse-power will vary from 9 to 12 lbs., for the sake of illustration, we will say 10 lbs., and that the engine is of such size as would require for a year's run $3,000 worth of coal; now, an ordinary adjustable cut-off engine with throttling governor, ought to save at least half that amount of coal, or say $1,500 per year; if the best automatic engine were employed using $2\frac{1}{2}$ lbs. of coal per horse-power, a further saving of $750 per year could be effected, or between the two extremes $2,250 per year in saving of coal, without interfering in any way with the power, with the exception, perhaps, that the automatic engine will furnish a better power than the former engine. It is easy to see that it is true economy to buy the best engine and pay the extra cost of construction, if the saving of fuel is an element entering into the question of selection.

The cost of an engine for any particular service is always to be taken into consideration, for it is possible to contract for a certain saving of coal at too high a price, not simply when paid out as the original purchase money, but with this economy of fuel, the purchaser may have many vexatious and damaging

delays caused by the breaking of the automatic mechanism of the engine. All such delays, which would not have occurred to an ordinary or simpler engine, are to be charged against any saving credited to the engine, which failed in producing a regular and constant power. Take a flouring mill for example, producing 400 barrels per day; it is easy to see how a single day's stoppage would interfere with the trade and shipment by the proprieters, yet it would require a very small break in an engine that would require less than a day for repairs.

This does not argue against high grade engines, but the purchaser should be certain that the engine when once on its foundations shall be as free from dangers of this kind as any other engine of similar economy.

There are engines, which, from their peculiar construction appear to be very complex, and this objection is often urged against them, while the fact is the complexity is apparent rather than real. Take the Corliss engine, for example; it is doubtful whether there is another automatic cut-off engine in successful use in this or any other country which has cost less for repairs during the last ten or twenty years. It is true it contains a great many separate pieces in the valve mechanism, but the pieces themselves are simple, durable, easily accessible and always in sight. These several parts are not liable to excessive wear, but such as there is can be readily adjusted.

The engines to be preferred are those in which the valve adjusting mechanism is outside of the steam chest and which is in plain sight at all times when the engine is in motion.

Location of engine. — This will depend upon circumstances, but it is far from true economy to place an engine in a dark cellar, or in some inconvenient place above ground. The engine as the prime mover, should have all the care and attention which may be needed to insure regular and efficient working.

Machinery in the dark is almost sure to be neglected. If the

design of the building, or the nature of the business, is such that the engine must be located underground, there should be some provision for letting in the daylight; the extra expense incurred will soon be saved by the order, cleanliness and fewer repairs required.

The engine should always be close to, but not in the boiler room. Many a high-priced engine has had its days of usefulness shortened by the abrasive action of fine ashes and coal dust coming in contact with the wearing surfaces. There should always be a wall or tight partition between the engine and fire room.

The foundations for an engine should be large and deep. Too many manufacturers in marking dimensions of foundation drawings for engines, make them altogether too shallow. The stability of an engine depends more on the depth than on the breadth of the foundations. Stone should be used for foundations rather than brick, but if the latter must be used they should be hard burned and laid in a good cement rather than a lime mortar. If the bottom of the pit dug for the engine foundation be wet, or the soil uncertain in its stability, it is a good plan to make a solid concrete block about a foot and a half thick, on which the foundation may be continued to the top. If such a concrete block be made with the right kind of cement it will be almost as hard and solid as a whole stone.

The most economical engine is the one in which high pressure steam can be used during such portion of the stroke as may be necessary, then quickly cut off by a valve, which shall not interfere with the exhaust at the opposite end of the cylinder, and allow the steam to expand in the cylinder to a pressure, which shall not fall below that necessary to overcome the back pressure on the piston. In general, the most successful cut-off engines use the boiler pressure for a distance of one-fifth to three-eighths of the stroke from the beginning; at this point the steam is cut off and allowed to expand throughout the balance of the stroke.

The gain by expansion consists in the admission of steam at a pressure much above the average required to do the work, and allowing it to follow but a small portion of the stroke, then expanding to a lower than the average pressure at the end of the stroke. The mean effective pressure on the piston is that by which the power of the engine is measured; hence, it follows that the higher economy is to be reached, other things being equal, where the mean effective pressure on the piston is highest when compared with the terminal pressure, or the pressure at the end of the stroke. In order to get this, a high initial pressure is used; the steam follows as short a distance as possible to keep the motion regular under a load, and then expanding down to as near the atmospheric pressure as possible.

The following table exhibits at a glance the performance of a non-condensing engine cutting off at different portions of the stroke. The initial pressure of steam being in each case eighty pounds per square inch.

CUT-OFF IN PARTS OF THE STROKE.

	$\frac{1}{10}$	$\frac{2}{10}$	$\frac{3}{10}$	$\frac{4}{10}$	$\frac{5}{10}$
Mean effective pressure .	18	35	48	57	65
Terminal pressure . . .	11	20	30	39	48
Pounds water per h'r per H. P.	27	24	25	27	28

Fractions are omitted in the above table and the nearest whole number given.

Governor. — Any automatic device by which the speed of an engine is controlled may properly be called a governor. There

are now two distinct methods by which the steam supplied to an engine is thus brought under control. The first is usually applied to slide valve engines having a fixed cut-off, and consists in the adjustment of a valve by which the *pressure* of steam in the cylinder is increased or diminished in order to maintain a constant rate of revolution with a variable load. The second device consists in a mechanism by which the whole boiler pressure is admitted to the cylinder, which is allowed to follow the piston to such portion of the stroke as will maintain a regular rate of revolution; the steam is then suddenly cut off at each half revolution of the engine, thus furnishing a greater or less *volume* of steam at a constant pressure. Neither of these two varieties of governors will act until a change in the rate of revolution of the engine occurs, and this change will either admit more or less steam as it is slower or faster than that for which the governor is adjusted. The commonest form of a governor consists of a vertical shaft to which are hinged two arms containing at their lower ends a ball of cast iron; as the shaft revolves the balls are carried outward by the action of what is commonly called centrifugal force; the greater the rate of revolution the farther will the balls be carried outward; advantage is taken of this property to regulate the admission of steam to the engine. The action of the balls and that of the valve include two distinct principles and should be considered separately; an excellent valve may be manipulated by an indifferent governor and so produce unsatisfactory results; on the other hand, the governor mechanism may be satisfactory in its operation, but being connected with a valve not properly balanced, is likely to cause a variable rate of revolution in the engine.

Fly-wheel. — The object in attaching a fly-wheel to an engine is to act as a moderator of speed. The action of the steam in the cylinder is variable throughout the stroke, against which the revolution of a heavy wheel acts as a constant resistance and limits the variations in speed by absorbing the surplus power of the first

portion of the stroke, and giving it out during the latter portion. The fly-wheel is simply a reservoir of power, it neither creates nor destroys it, and the only reason why it is attached to an engine is to simply regulate the speed between certain permitted variations, which are necessary to cause the governor to act, and to equalize the rate of revolution for all portions of the stroke, thus converting a variable reciprocating motion into a constant rotary one. It is considered good practice to make the diameter of the fly-wheel four times the length of the stroke for ordinary engines, in which the stroke is equal to twice the diameter of the cylinder. This may be taken as a fair proportion in engine building, and furnishes a wheel sufficiently large to equalize the strain and reduce any variation in speed to within very narrow limits, if the engine is supplied with a proper governor. The greater the number of revolutions at which the engine runs, the smaller in diameter may be the fly-wheel, and it may also be largely reduced in weight for engines developing the same power.

Horse-power. — By this term is meant 33,000 pounds raised one foot high in one minute. The horse-power of an engine may be found by multiplying the area of the piston in square inches by the mean effective pressure; this will give the total pressure on the piston; multiply this total pressure by the length of the stroke of the piston in feet; this will give the work done in one stroke of the piston; multiply this product by the number of strokes the piston makes per minute, which will give the total work done by the steam in one minute; to get the horse-power, divide this last product by 33,000. From this deduct, say, 20 per cent, for various losses, such as friction, condensation, leakage, etc.

CARE AND MANAGEMENT OF A STEAM ENGINE.

It is to be supposed to begin with that the engine is correctly designed and well made, and that, after a suitable selection of an

engine for the work to be done, nothing now remains except proper care and management.

Lubrication. — The first and all-important thing in regard to keeping an engine in good working order is to see that it is properly lubricated. This does not imply, neither is it intended to encourage, the use of oil to excess; all that is needed is simply a film of oil between the wearing surfaces. It is marvelous how small a quantity of oil is required when of good quality and continuously applied. There are several self-feeding lubricators in the market which have been tested for years and are a pronounced success; these include crank-pin oilers, in which the oscillatory motion of the oil makes a very efficient self-feeding device, the flow being regulated by means of an adjustable opening to the crank-pin, or in the adjustment of a valve by which its lift is regulated by each throw of the crank; and in others by a continual flow through a suitable tube containing a wick or other porous substance. For stationary engines, it is desirable that the main body of the oiler be made of glass that the flow of oil may be closely watched and adjusted accordingly. For the reciprocating and rotary parts of the engine, a modification of the above mentioned oilers may be used. They are of various patterns and devices and many of them very good. It is also a good plan to have some device by which the cross-head at each end of each stroke will take up and carry with it a certain amount of oil; for the lower half of the slide this is not difficult to arrange; for the upper side an automatic feeder placed in the middle of the slides will provide ample lubrication.

For oiling the main bearing there should be two separate devices, one an automatic glass oiler; and in addition, a large tallow cup attached to the cap of the bearing. This cup should be filled with tallow mixed with powdered plumbago; the openings from the bottom of the cup to the shaft should be not less than quarter-inch for small engines, and three-eighths to half-inch

for larger ones; so long as the main bearing runs cool the tallow will remain in the cup unmelted; but if heating begins, the tallow will melt and run down on the surface of the revolving shaft, and thus provide an efficient remedy when needed. For oiling the valves and piston, a self-feeding lubricator should be attached to the steam pipe; this by a continuous flow of oil will be found not only satisfactory in its practical working, but economical in the use of oil.

In selecting an oil for an engine, it is in general better to use a mineral rather than an animal oil, especially for use in the valve chest and cylinder. The objection to an animal oil, and especially to tallow or suet, is that it decomposes by the action of heat, often coating the surface of the steam chest, the piston ends and the cylinder heads with a deposit of hard fatty matter; or forms into small balls not unlike shoemaker's wax. There is no such decomposition and formation in connection with mineral oils, which may now be had of uniform quality and consistency, and at much lower prices than animal oils.

The slide valve should be kept properly set and should be examined occasionally to see that the face and seat are in good condition. So long as this is the case, the valve mechanism and the valve itself must be let alone and not tampered with.

The piston packing will need looking after occasionally to see that it does not gum up and stick fast, which it is very likely to do when the cylinder is lubricated with tallow or animal oil.

The rings should fit the cylinder snugly and should be under as little tension as possible and insure perfect contact. If the rings are set out too tight they are liable to scratch or cut the cylinder; if too loose, the steam will blow through from one end of the cylinder, past the piston and into the other. In adjusting the springs in the piston, care must be exercised that the adjustments are such as will keep the piston rod exactly central, to prevent springing the rod, or causing excessive wear on the stuf-

fing-box. There are several packings, which do not require this adjustment, the rings being narrow, and either expanding by their own tension or by means of springs underneath. The only thing to be done with such a packing is to keep it clean, and when lubricated with a mineral oil this is not a difficult matter. If it groans, take rings out and file sharp edges off.

The stuffing-boxes whether for the piston or valve-stem need to be looked after carefully, and to prevent leaking, will require tightening from time to time. There are several kinds of ready-made packings in the market, containing rubber, canvas, garlock, soapstone, asbestos and other substances which form the basis of a good durable packing. These can be had in sizes suitable for all ordinary purposes, and their use is recommended. In the absence of any of these, a packing made of clean manila or hemp fiber will serve a useful purpose. Formerly it was the only substance used, but is being gradually superseded by the other kinds mentioned above. In packing the small and delicate parts, such as a governor stem, a good packing is made by pleating together three or more strands of cotton candle-wick. This is soft, pliable, free from anything like grit, and will not get hard until soaked with grease and baked into a brittle fiberless substance not easily described.

Crank-pins. — There are few things more troublesome to an engineer than a hot crank-pin, and it is sometimes very difficult to get at the real reason why it heats. Among the principal reasons for heating are: the main shaft is not "square" with the engine, or, that the pin is not properly fitted to the crank; or, perhaps, it is too small in diameter — defects which are to be remedied as soon as practicable. Heating is often caused by the boxes being keyed too tightly, or by insufficient lubrication. There are now several good self-feeding lubricators in the market which will supply the oil to a crank-pin continuously; these are recommended rather than the old style of oil cup, which was

not only uncertain, but doubtful in its action. Many troublesome crank-pins have been cured of heating by this simple matter of constant lubrication. When the crank-pin is rather small for the engine and the load variable, there is a possibility of having a hot pin at any time; it is advisable to have ready some simple and effective expedient to be applied when it does occur; for this there is perhaps nothing better and safer than a mixture of good lard oil and sulphur.

Connecting rod brasses. — In quick running engines the brasses should be fitted metal to metal; or, if this is not desirable, several strips of tin or sheet brass should be inserted between them and keyed up tight. This gives a rigidity to a joint which is difficult to secure when the brasses have a certain amount of play in the strap. It is a common practice to bore the brasses slightly larger than the pin, so that when fitted to it the hole shall be slightly oval, and thus permit a freer lubrication than is secured by a close fit around the whole circumference.

Knocking. — There are several causes which, combined or singly, tend to produce knocking in steam engines. In most cases the difficulty will be found to be in the connecting rod brasses; but whether in the crank-pin end or at the cross-head is not easily determined in all cases. A very slight motion will often produce a very disagreeable noise; the remedy is, in most cases, very simple, and consists in simply tightening the brasses by means of the key or other device that may have been provided for their adjustment. In adjusting a key it is the common practice to drive it down as far as it will go, marking with a knife blade the upper edge of the strap, then drive the key back until it is loose; after which drive it down again, until the line scratched on the key is within $\frac{1}{4}$ or $\frac{1}{8}$ inch of the top of the strap. The size of the strap joint and the judgment of the person in charge must decide the best distance. This may be done

at both ends of the connecting rod. On starting the engine, the cross-head and crank-pin must be carefully watched, and upon the slightest indication of heating, the engine should be stopped and the key driven back a little farther. A slight warmth is not particularly objectionable, and will, as a general thing, correct itself after a short run. Knocking is sometimes occasioned by a misfit, either in the piston, or cross-head and the piston-rod. These connections should be carefully examined, and under no circumstances should lost motion be permitted at either end of the piston rod.

If the means of securing are such that the person in charge can properly fasten the piston to the rod, he should see that it is kept tight; if not, then it should be sent to the repair shop at once, as there is no telling when an accident is likely to overtake an engine with a loose piston.

The connection between the piston-rod and cross-head is usually fitted with a key and furnishes a ready means of tightening the joint, if proper allowance has been made for the draft of the key. In case there has not, the piston-rod and cross-head should be filed out so that the draft of the key will insure a good tight joint when driven down.

The main bearing should be examined and if there should be too much lateral movement of the shaft, the side boxes might then be adjusted until the shaft turns freely, but has no motion other than a rotary one. The cap to the main bearing should also be carefully examined, as it may need screwing down and thus prevent an upward movement of the shaft at each stroke; this applies more particularly to quick running engines.

Engines which have been in use for some time are likely to have a knock caused by the piston striking the head. This is brought about by having a very small clearance in the cylinder and in not providing, by suitable liners, for the wear of the connecting rod brasses. In a case of this kind, liners should be inserted behind

the brasses in the connecting rod, or new brasses put in, which will restore the piston to its original position.

Knocking may be caused by defects in the construction of the engine; such, for example, as not being in line, the crank-pin not at right angles to the crank, the shaft may be out of line, etc.

Whenever the cause is one in which it can be shown that it is a constructive defect, there is but one remedy, and that is the replacing of that part, or the assembling of the whole until perfect truth is had in alignment of all the parts. This will require the services of an experienced engineer but all improperly fitting pieces should be replaced by new ones as a safeguard against accident, which is likely sooner or later to overtake badly fitting pieces.

If the boiler is furnishing wet steam, or priming, so as to force water into the steam pipe, it will collect in the cylinder and will not only cause knocking, but on account of its being practically incompressible there is danger of knocking out a cylinder head, bending the piston-rod, or doing other damage to the engine. The cylinder cocks should be opened to drain any collected water away from the cylinder.

Repairs. — Whenever it is necessary to make repairs the work should be done at once; oftentimes a single day's delay will increase the extent and cost fourfold. If an engine is properly designed and built, the repairs required ought to be very trivial for the first few years it is run, if it has had proper care. It may be said in reply to this "true, but accidents will happen in spite of every care and precaution." That accidents do occur is true enough; that they occur in spite of every care and precaution is not true. In almost every case, accidents may be traced directly back to either a want of care, negligence, or to a mistake.

Fitting slide-valves. — The practice of fitting a slide-valve to its seat by grinding both together with oil and emery, is wrong and should never be resorted to. The proper way to fit the sur-

faces is by scraping; this insures a more accurate bearing to begin with, and will also be entirely free from the fine grains of emery which find their way and become imbedded in the pores of the casting, and are thus liable to cut the valve face and destroy its accuracy. The scraping of the valve and seat has a beneficial effect by causing the removal of the fine particles of iron, which are loosened by the action of the cutting tool in the planing machine, and which ought to be fully removed before the engine leaves the manufacturers' hands. Aside from this, it is doubtful whether the scraping amounts to anything practically, for the reason that the cylinder and valve are fitted cold, and their relative positions are distorted by the action of the heat of the steam, once the engine is in use. The scraping, which simply renders the valve face and seat smooth and hard, is all that is sufficient to begin with, and may be re-scraped after the valve has been in use a few days, should it be found necessary, which will not often be the case in small and ordinary sized engines.

Eccentric straps are likely to need repairs as soon as anything about an engine. They should be carefully watched at all times. If they are likely to run hot, it is also probable there is more or less abrasion or cutting going on, and if prompt measures are not taken to arrest it, they are likely to cut fast to the eccentric, and a breakage is sure to occur.

When the straps begin to heat, the bolts should be slackened a little, and at night, or perhaps at noon, the straps should be taken off and all cuttings carefully removed with a scraper (not with a file); the rough surfaces on the eccentric should be removed in the same manner.

The straps should be run loose for a few days, gradually tightening as a good wearing surface is obtained.

The main bearing, if neglected, is a very troublesome journal to keep in order. The repairs generally needed are those which

attend overheating and cutting. The shaft, whenever possible, should be lifted out of the bearing, and both the shaft, bottom of main bearing and side boxes, carefully scraped and made perfectly smooth. It sometimes occurs that small beads of metal project above the surface of the shaft which are often so hard that neither a scraper nor file will remove them; chipping is then resorted to and the fitting completed with a file and fine emery cloth.

Heating of journals. — A very common cause for the heating of journals having brasses and boxes composed of two halves, is that both halves alter their shape from causes attending their wear. Thus, most engineers will have noticed that, although there is no wear between the sides of a brass and the jaws of a box, yet in time the brass becomes a loose fit in the box. Now, since the sides of the brass have, when fitted, no movement in the box, it is evident that this cannot have proceeded from wear between those surfaces, and it remains to find what causes this looseness. Most engineers will also have observed that though the bottom or bedding surfaces of a brass and of the box may have been carefully filed to fit each other when new, yet if in the course of time the brasses be taken out and examined, and more especially the bottom brass that receives the weight, the file marks will become effaced on all parts where the surfaces have bedded together well, the surface having a dull bronze and condensed appearance. This is caused by the vibrations under pressure having condensed the metal. Now, this condensation of the metal moves or stretches it, and causes the sides of the brass to move away from the sides of the box, and, consequently, to close upon the journal, creating excessive friction that may often, and very often does, cause heating. It is for this reason that on such brasses the sides of the brass boxes are, by a majority of engineers, eased away at and near the joint, and it follows from this cause the same easing away is a remedy.

Governor. — It not infrequently occurs that after an ordinary

throttling engine has been used a few years, the speed becomes variable to such a degree that it interferes with the proper running of the machinery. This occurrence can generally be traced directly to the governor. When it does occur, the governor should be taken apart and thoroughly examined; if the needed repairs are such as can be easily made in an ordinary repair shop, they should be made at once; if not, a new governor should be purchased. The price of governors is now so low that it is better and more economical to buy a new one than lose the time and pay the bills for repairing an old one.

AUTOMATIC ENGINES.

In the care and management of this class of engines, it is difficult to say just what particular attention they need, owing to the variety of styles and the peculiarities of each. As a rule, however, they require first, to be kept well oiled; second, to be kept clean; third, to be kept well packed; and fourth, to be let alone nights and Sundays. There is little doubt that there has been more direct loss resulting from a ceaseless tinkering with an engine than results from legitimate wear and tear to which the engine is subjected. It is not to be inferred from the preceding remark that builders of this class of engines are infallible; it might be difficult to prove any such assertion in case it was made; but it may be said with truth, that the engines of this class now in the market are carefully designed, well proportioned, of good materials and workmanship, and as examples of mechanism are entitled to take very high rank. Engineers know of several engines of this class which have not cost their owners for repairs so much as five dollars in five years' constant use. It is essential to the economical working of these engines that the cut-off mechanism be in good order and properly adjusted. Whenever the valves need resetting, the final adjustment should be made

with a load on the engine and with the indicator attached to the cylinder, the valves being set by the card rather than by the eye. No general rule can be given for setting the valves, as the practice varies with the size and speed of the engine; nor is any rule needed, for the indicator will furnish all the data required. The adjustments may then be made so as to secure prompt admission, sharp cut-off, prompt release, and the proper compression.

TO FIND THE DEAD CENTERS.

When setting the valve of an engine by measuring the lead, as is the usual method, it is necessary that the crank be accurately placed on the dead centers at each end of the stroke. Sometimes an engineer, when adjusting the valves of his engine, will attempt to place the crank on the dead center by watching for the point at which the travel of the cross-head stops, or by the appearance of the connecting-rod as related to the crank. These methods are totally unreliable for obtaining accurate results, especially the first one mentioned. The travel of the cross-head and the piston near the point of reversal of motion is very slow when compared with the valve. The velocity of travel of the valve is at nearly its maximum amount when the crank is on the dead center, and a slight error in finding the dead center point makes a very appreciable error in the position of the valve, with a subsequent error in its proper setting.

There are several methods for finding the dead center. The method that can be recommended and the one that should always be used when the dead center of an engine is to be found is that familiarly known as "tramming." The dead centers when found by this method, are geometrically accurate, no matter if the engine is out of level or if the shaft is above or below the axis of the cylinder. Some simple tools are required which are generally available, with the exception of the trams, which may be readily

made for the purpose. Two trams are required, one of which should be 6″ or 7″ long and the other about 24″ or 30″, as the condition may require. The smaller tram may be made of $\frac{1}{4}$″ steel wire with the points turned over at right angles to the body, so as to project about 1″. The points should be sharpened so that a hair line may be drawn by them. The larger tram should be made from rod of at least $\frac{3}{8}$″ diameter and the points made in the same way as for the smaller tram. Oftentimes, the long tram

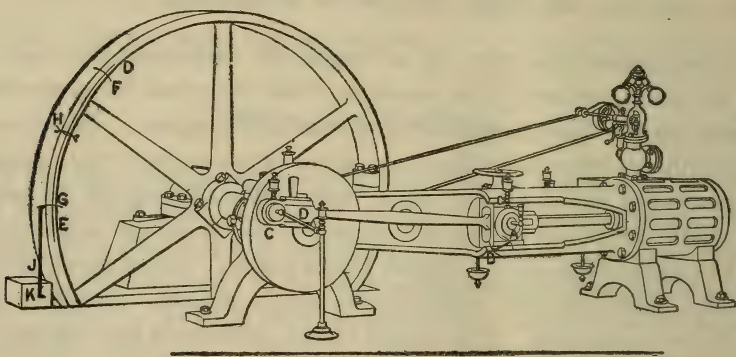


Fig. 130. Finding the dead center.

is made with one leg longer than the other, on account of being handier to reach some stationary part, but this is a minor point, which has nothing to do with the principle to be described. The other tools required are a light hammer, a prick-punch, a pair of 10″ or 12″ wing dividers and a hermaphrodite caliper, or a scribing block. A piece of chalk will also be found convenient to facilitate scribing lines on the metal parts with the trams or dividers. Fig. 130 shows the use of the trams.

Having the necessary tools, we are ready to begin operations,—and may start at either end of the stroke, as circumstances may favor. The fly-wheel is turned so that the crank stands at about the angle shown in the accompanying illustration, which

may, however, be approximated as the operator may desire. The effort made, being to give sweep enough to the cross-head to allow accurate measurements and still not have such an excessive arc on the fly-wheel as to make its bisection difficult.

A prick mark is made on the guides, or some convenient stationary point, as at *B*, and an arc struck on the cross-head with the small tram. At the same time, an arc is scribed on the rim of the fly-wheel at *G*, using some convenient point for the lower point of the tram as at *K*. The fly-wheel is now turned until the crank passes the center and the cross-head travels back until the scribed line will coincide exactly with the point of the tram when held in the same position as in the first case. When this point has been reached, the wheel is stopped and a second arc is scribed on the fly-wheel rim at *F* with the tram *J*. The hermaphrodite caliper, or the scribing block, is now used to scribe a concentric line *D E* on the fly-wheel rim and the arc *C F* is bisected with the dividers. When the center *H* has been accurately located, it should be carefully prick-marked. The scribing of the concentric line *D E* is a refinement that is not strictly necessary if care be taken to locate the points of the dividers at the same distance from the outer periphery of the wheel in each instance when finding the center *H*. The marks left by the lathe tool will sometimes be plain enough for a guide. When the center *H* has been found, the fly-wheel is turned so that the point of the tram will fall into the prick-mark *H* when its lower end is in the stationary point *K*. When this condition is effected, the crank is exactly on the dead center and the position of the valve may be taken with confidence that its location at the dead center point is accurately found. The same procedure is followed to place the crank on the dead center at the opposite end of the stroke.

The cut on page 198 is an elevation of Tandem Compound Engine, showing engine erected on brick foundation. It also shows a line through cylinders; also a line over the shaft.

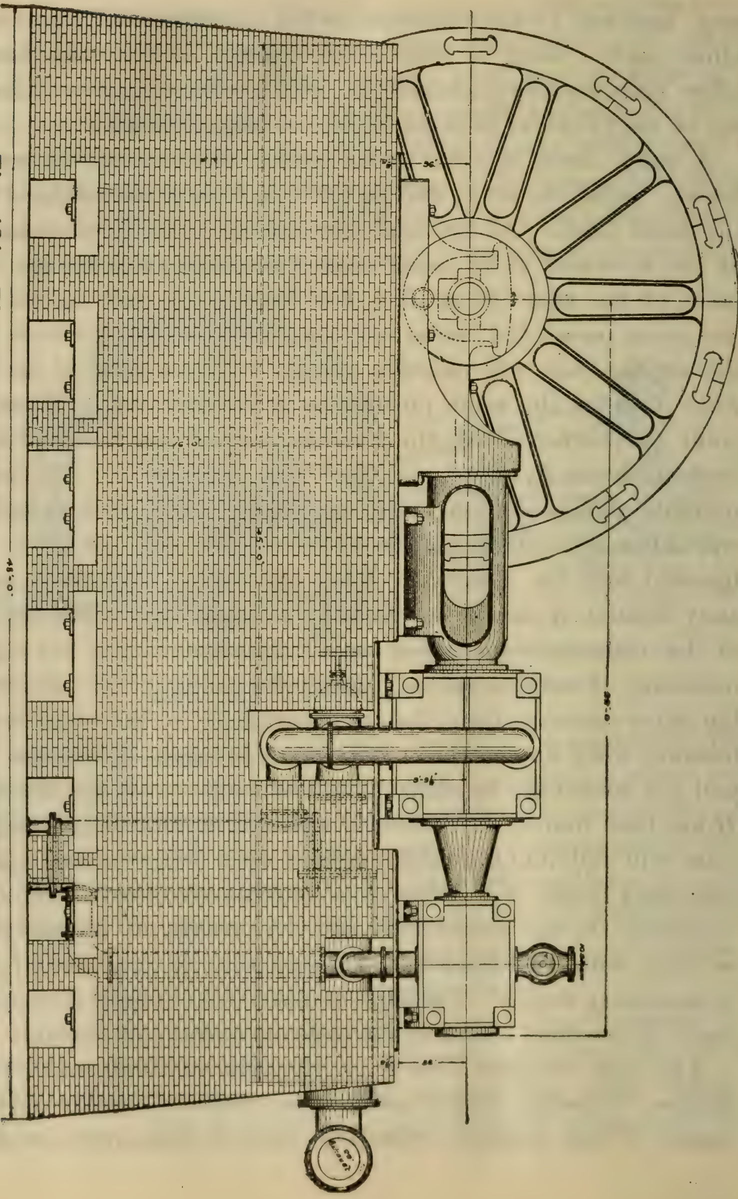


Fig. 131. Elevation of tandem compound engine, showing engine erected on brick foundation.

These lines are used in the erection of a new engine, or to line up an old one, or with an engine that is out of line. The cut also shows how the foundation is made; also how the anchor bolt is fastened.

The cut on page 200 shows how to pipe a Twin Tandem Compound Condensing Engine. The plan shows two receivers, heaters, relief valves, gate valves, etc., and is so arranged that either side can be run independently of the other. It also shows how to line a pair of these engines by following the lines and noting the distance between each line. An engineer would have no trouble in lining up a pair of these engines.

HOW TO LINE AN ENGINE.

The method followed when lining different types of engines, such as vertical, horizontal, portable, etc. is as follows: —

The method followed in lining any piston engine is essentially the same in all cases, as far as determining when adjustments are needed. The method of making the adjustments after the character and amount of them is determined, depends entirely on the construction of the engine, and will necessarily have to be determined in each individual case. Lining an engine consists of adjusting the guides so they shall be parallel to the bore of the cylinder, and in such a position that the center of the piston socket of the cross-head shall coincide with the axis of the cylinder. Under these conditions only, can the piston and cross-head travel through the stroke freely, and without distorting any of the parts. After this adjustment has been made, the truth of the right-angle position of the shaft must be determined as being "out of square;" this will make an engine run badly, and is often the unsuspected cause of much trouble to engineers. We will assume that we have an engine with four-bar or locomotive guides, and that the connecting rod, cross-head, back cylinder

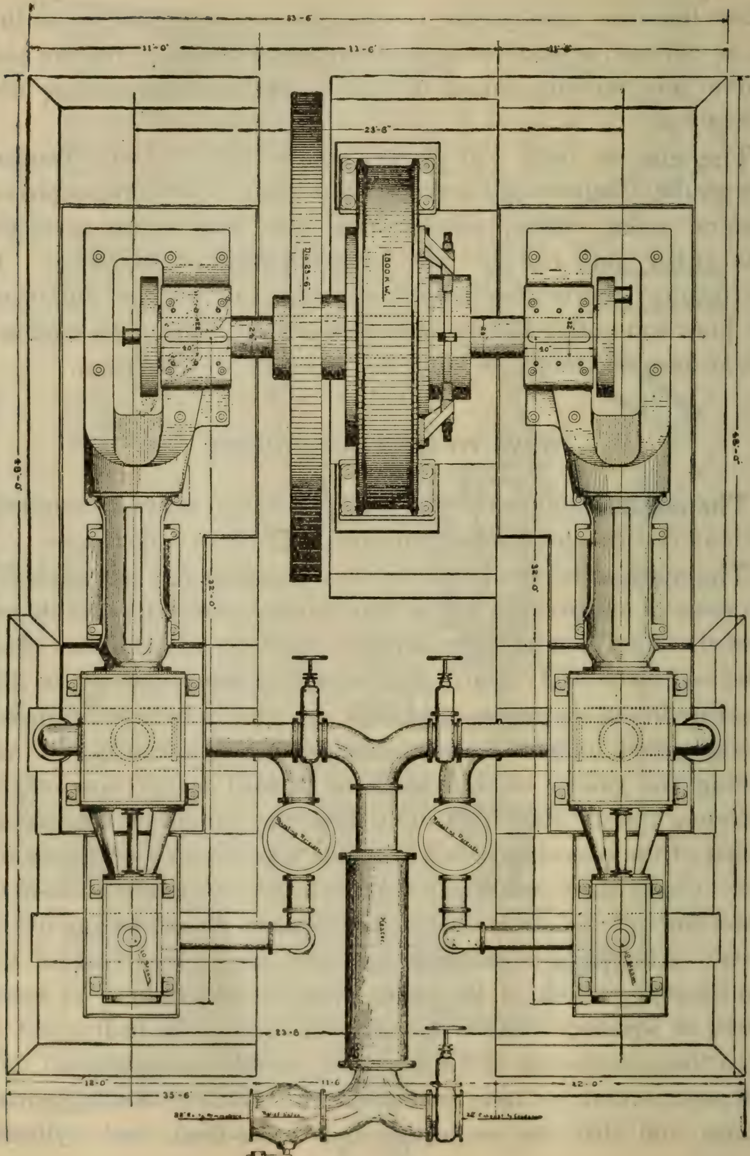


Fig. 132. Plan of twin tandem compound engine.

head and piston have been removed. If the engine is of the horizontal type, the first step will properly be to ascertain if the engine is level on the foundation, and if not, proceed to make it so. After having leveled the engine, stretch a smooth linen line, as shown in Fig. 133, through the bore of the cylinder and the stuffing-box, to a point beyond the shaft, where it should be attached to an iron rod driven into the floor. The other end is fastened to a cross-bar bolted across the face of the cylinder to

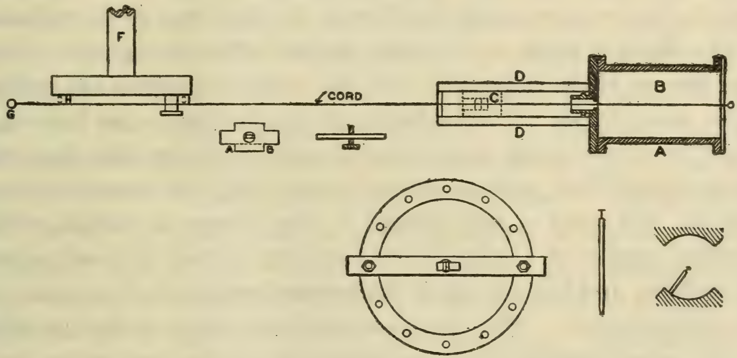


Fig. 133. Lining up an engine.

two of the studs, as shown in Fig. 133, or the bar may preferably be somewhat longer than one-half of the diameter of the cylinder, and with a saw cut for a short distance lengthwise at the inner end. In this case, it is held by only one of the cylinder studs and can be somewhat more easily adjusted. The line or cord is adjusted to approximately the proper position, and is drawn taut and fastened through the cross-bar by being tied to a short stick that is too long to pass through the hole. In this position it is held by the friction, and can be readily adjusted to the required position. An assistant is required to move the line in the directions indicated, as the work proceeds, and then we are ready to center it in the cylinder. The only tool required for this purpose is a light pine stick of slightly less length than the radius of the

bore, and it should have an ordinary pin pushed into the head for a "feeler." Now adjust the line in the cylinder so that the head of the pin will just tick the line from four points of the counterbore, which is always the part of the cylinder to work from, as it is not affected by the wear. The line should then be adjusted to the center of the other end of the cylinder, but not from the stuffing-box, as this is likely to be out of center somewhat. Make the adjustment at this end from the counterbore, if possible, the same as in the first instance, and then it will be necessary to try the position of the line in the back end of the cylinder, as the changes made at the other end will affect it slightly. After the line is truly centered, we are ready to adjust the guides. With some types of cross-heads, it is possible to use the cross-head for determining the proper location of the guides, but with the ordinary form, such as shown in Fig. 133, this cannot be done, but we will need a tool similar to that shown in sketch, which consists simply of a piece of flat iron long enough to reach across the guides, and having a hole drilled and tapped in the center for the thumb-screw. This thumb-screw is adjusted so that its point is the same distance from the lower side of the bar, as the lower face of the wings of the cross-head are from the center of the piston socket. To find this distance, lay a straight edge across the end of the cross-head and draw the line *A B*, and then, having found the center of the hole, the measurement may be accurately taken. The lower guides are now adjusted by the tool, so that the point of the screw will tick the line throughout the length, and then the top guides are put in position with the cross-head in place and adjusted for a proper working fit.

Before removing the line from the cylinder, however, the shaft should be tested for the truth of its right-angle position, which may be done by calipering between the crank disc and the line at the points *H* and *I*. If the distances are equal, the shaft is square with the bore of the cylinder, providing, of course, that

the disc is faced true with the shaft. If there is any doubt as to its accuracy, turn the shaft as nearly half way around as the crank-pin will admit without disturbing the line. Then caliper the distance of a point on the disc that will not be far removed from the first position, thus reducing the chance for error. If the shaft shows "out," move the outward bearing until the measurements show equal in both positions. The horizontal truth of the shaft can be found by laying a level on it, and if "out," raise or lower the out-board bearing until the level shows fair. Work of this kind requires skill and patience and belongs properly to the sphere of the chief engineer. It requires a delicacy of touch and an appreciation of what is meant by close measurement that can come only through experience. In centering the line, one should be able to detect when it is as little as $\frac{1}{1000}$ of an inch out of center. A piece of ordinary tissue paper is about .00125 inch thick. A man should be able, therefore, to adjust a line so accurately that if the "feeler," with one or more pieces of the paper under it, just clips the line, it will miss the line when one thickness is removed. While it may not always be necessary to work as closely as this, a person cannot expect to line up engines successfully until he has a full knowledge of what this degree of accuracy means.

Engine Formulas.

$$\text{Diam. cyl. for given H.P.} = \sqrt{\frac{33000 \times \text{H.P.}}{\text{Piston speed} \times \text{M.E.P.}} \div .7854}$$

$$\text{Stroke in feet} = \frac{\text{Piston speed in feet per min.}}{\text{Revs.} \times 2}$$

$$\text{Revs. per min.} = \frac{\text{Piston speed in feet per min.}}{\text{Length of stroke in feet} \times 2}$$

$$\text{Piston speed} = \frac{33000 \times \text{H.P.}}{\text{Area of Piston in sq. inches} \times \text{M.E.P.}}$$

$$\text{Area of piston in sq. ins.} = \frac{33000 \times \text{H.P.}}{\text{M.E.P.} \times \text{Piston speed}}$$

$$\text{M.E.P. required} = \frac{33000 \times \text{H.P.}}{\text{Area of Piston} \times \text{Piston speed}}$$

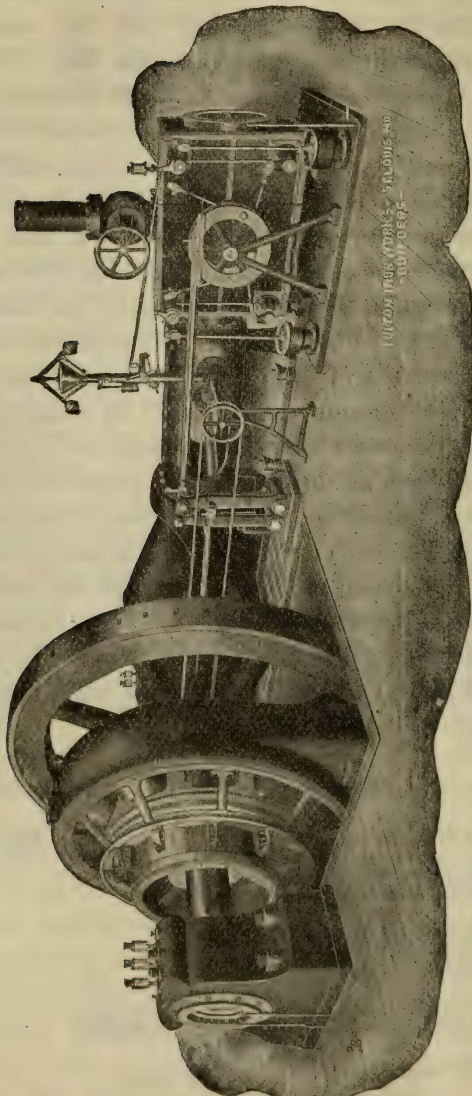


Fig. 134. Rear view of double eccentric Corliss engine.

CHAPTER XIa.

DIRECTIONS FOR SETTING UP, ADJUSTING AND RUNNING THE CORLISS STEAM ENGINE.

Location of foundation. — The foundation must be at right angles with main line shaft. If main line shaft is not already in position, then foundation must be set by two points, located and connected with a line parallel with the buildings, and at right angles to an imaginary line through center of cylinder.

Foundation plans should show all center lines. If a templet is furnished to locate the foundation accurately for the mason, the center line of engine cylinder and guides and right angle for crank center are drawn thereon.

Cap Stones. — Examine carefully the lap faces of cap stones and, if necessary, have them trimmed off by cutter or mason, so that each is true and level, and in exactly the plane shown in foundation plan.

Cylinders and frame. — Put engine cylinder and frame in position and bolt them together.

Lining off crank shaft and out-end bearing. — Stretch a line at right angles to main center line, through main bearing to represent center line of crank shaft. See that this line is exactly in the center and level. By this line place out-end bearing square and true. Put crank shaft in its bearings after bottom box has been placed in main bearings. Insert quarter boxes and adjusting wedges into main bearing and put cap on.

To ascertain that shaft is at exact right angles to main center line, turn engine shaft until the crank pin comes nearly to the main

center line, then with a pair of calipers, or rule, measure from shoulder of crank-pin to line, and after noting this distance, turn the crank back towards opposite center until pin is in same relative position to line, and measure again. If both measurements do not correspond, out-end bearing must be moved either way as required, until measurements show equal. Then take up slack around shaft in main bearing, being careful not to force the adjusting wedge too tight.

Fly-wheels. — The fly-wheel is next placed on shaft and firmly keyed in position.

Placing valve gear. — Steam and exhaust valve covers or bonnets on valve gear side are next bolted to place, taking care that no dirt or foreign substance gets between the surface underneath the covers.

Valve stems are inserted from opposite or front of cylinder and the valves put in after them, the *T* head of valve stem entering slot in valve. Couple up all valve gear parts, i. e., disc plate, valve-stem cranks, valve-connecting rods, dash-pots and dash-pot rods, valve-rod rocker, eccentric and straps on crank-shaft, first and second eccentric rods. The dash-pots should be thoroughly cleaned and oiled before putting in place.

ADJUSTMENT OF CORLISS VALVE GEAR WITH SINGLE AND DOUBLE ECCENTRICS.

A brief description of the essential parts of the Corliss engine valve gear will assist in obtaining a clear conception of the subject.

When a single eccentric drives both steam and exhaust valves the range of cut-off is limited to about half the piston stroke. This will become obvious by considering the following necessary conditions: —

After the eccentric has reached the extreme of its throw as shown in Fig. 135 in either direction all valve gear motions are reversed.

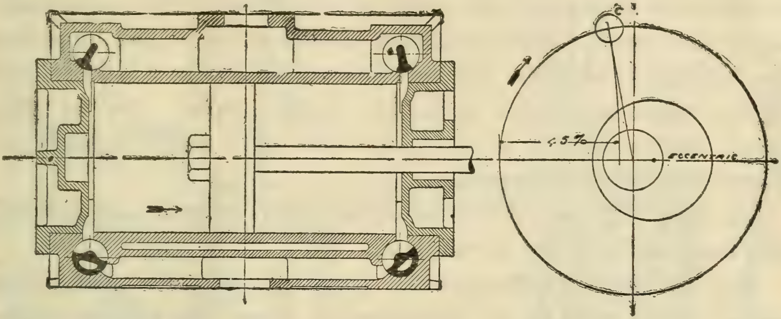


Fig. 135. Showing eccentric in extreme position.

The steam valve must be released before the eccentric motion is reversed, for if the hook does not strike the knock-off cam during its forward motion, it cannot strike it during its return motion.

The maximum exhaust opening, or the middle of the exhaust period, must occur when the eccentric is at the extreme of its throw as in Fig. 135.

Now, in order to release the expanded steam in the cylinder before the commencement of the return stroke and to secure the exhaust closure a little before the end of the return stroke, the middle of the exhaust period or the extreme of the eccentric throw must evidently occur before the middle of the return stroke, and, therefore, the extreme throw of the eccentric in the opposite direction must occur before the middle of the forward stroke, and the valve must be released before this point is reached if released at all.

It will be understood from the foregoing that late release and late exhaust closures are conditions imposed by the single eccentric valve gear, and these conditions agree very well with moderate rotative speed; but at higher speed earlier release and

more compression may be required. This may be effected by moving the eccentric forward on the shaft, but the reversing of the steam valve motion would then occur at an earlier stage of the forward stroke and the range of cut-off would be correspondingly shortened. Earlier exhaust closure could be had by giving the exhaust valve more lap, but this would involve a later release of the expanded steam at the end of the stroke. On the other hand, shortening the exhaust lap would give earlier release but insufficient or no compression.

In Figs. 136 and 137 similar capital letters of reference indicate the same parts of the mechanism.

Fig. 136 shows all the essential parts of the valve gear. The steam valves work in the chambers *S S* and the exhaust valves work in the chambers *E E*. The double-armed levers *D D* work loosely on the hubs of the steam bonnets; they are connected to the wrist-plate *B* by the rods *K K*, the levers *M M* are keyed to the valve stems *J J*, and are also connected by the rods *O O* to the dash-pots *P P*. The double-armed levers *D* carry at their outer ends what are called steam hooks, *F F*, these being provided with hardened steel catch plates, which engage with arms *M M*, making the arm *M* and the hook *F* work in unison until steam is to be cut off. At this point another set of levers or cams *G G*, which are connected by the cam rods *H H*, to the governor, come into play, causing the catch plates on the hooks *F* to release the arms *M M*, the outer ends of which are then pulled downwards by the dash-pot plunger, causing the steam valves to rotate on their axis and thus cut off steam. These are the essential features of the Corliss gear.

The exhaust valve arms *N* are connected to the wrist-plate by the rods *L L*, and it is seen that all the valves receive their motion from the wrist-plate *B;* the latter receives its motion from the hook-rod *A;* this rod is generally attached to a rocker arm not shown; to this arm the eccentric rod is

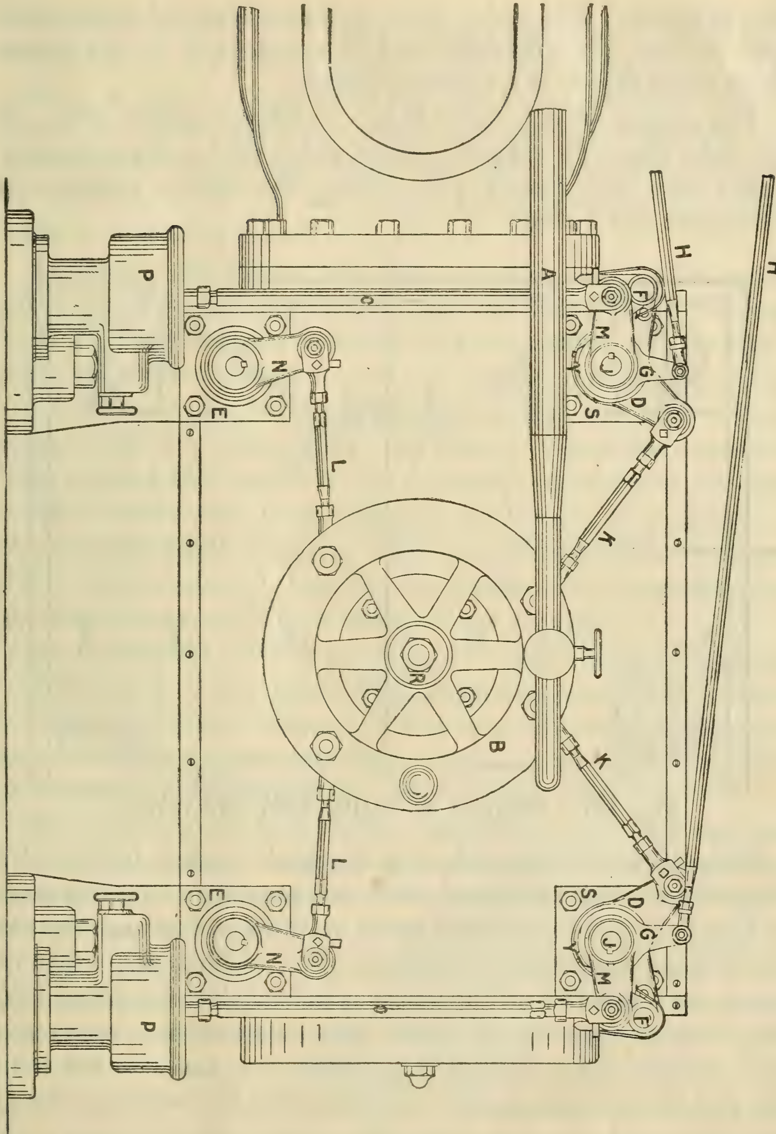


Fig. 136. Side view of Corliss valve gear.

also attached. The rocker arm is usually placed about midway between the wrist-plate and eccentric, and in the center of its travel stands in a vertical position.

The setting of the valves is not a difficult matter, when, on the wrist-plate, its support, valves and cylinder, the customary marks have been placed for finding the relative positions of wrist-plate and valves.

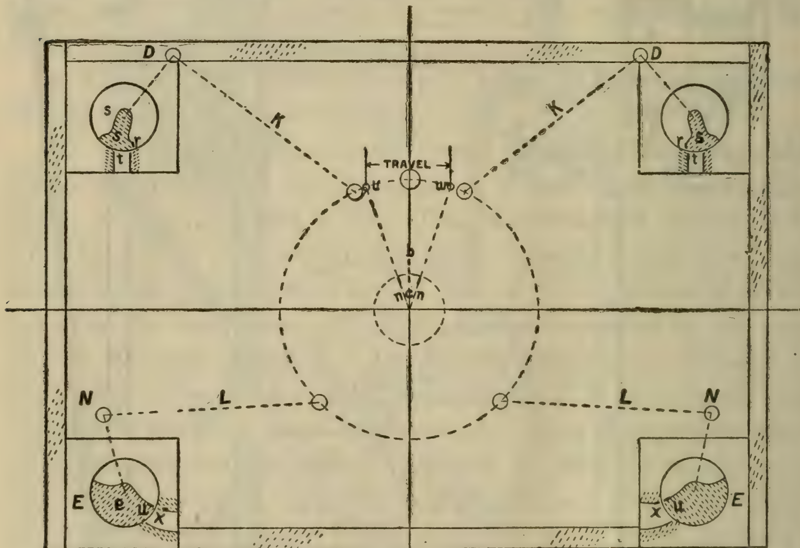


Fig. 137. Diagram of Corliss gear and valves.

Now, referring to Fig. 137, when the back bonnets of the valve chambers have been taken off, there will generally be found a mark or line, *r*, on the end of each steam valve *s s*, coinciding with the working or opening edge of each valve; another line, *t*, will be found on each face of the steam valve chamber coinciding with the working edge of the steam port. The exhaust valves and their chambers are marked in a similar way, i. e., the line *u* on the end of each exhaust valve coincides with the working edge of the valve, and the line *x*, on the face of each exhaust valve

chamber, coincides with the working edge of the exhaust port. On the hub of the wrist-plate will be found three lines n, c, n, placed in such a way that when the line c coincides with the line b on wrist-plate, the wrist-plate will stand exactly in the center of its motion, and when the line b coincides with either of the lines n, n, the wrist-plate will be at one of the extreme ends v or w of its travel.

In setting the valves, the first step will be to set the wrist-plate in its central position, so that the lines b and c will coincide, and fasten the wrist-plate in this position by placing a piece of paper between it and the washer R on its supporting pin. Now set the steam valves so that they will have a slight amount of lap, that is to say, the lines r, r, must have moved a little beyond the lines t, t. The amount of this lap depends much on individual preference and experience; it ranges from $\frac{1}{16}$ to $\frac{1}{4}$ for small engines, and from $\frac{1}{4}$ to $\frac{3}{8}$ inch for comparatively large engines. This lap is obtained by lengthening or shortening the rods K K by means of the adjusting nuts.

Now place the exhaust valves e, e, by lengthening or shortening the rods L L by means of the adjusting nuts, in a position so that the working edges will just open the exhaust ports, or, in other words, place the lines u and x in line with each other as indicated in illustration.

The next step will be to adjust the rocker arm. Set this arm in a vertical position by means of a plumb line, and connect the eccentric rod to it; then turn the eccentric around on the shaft, and see that the extreme points of travel are at equal distances from the plumb line. To secure this a little adjustment in the stub end of the eccentric rod may be necessary. Now connect the hook rod A to the wrist-plate. The paper between the wrist-plate and the washer on the supporting pin should now be taken out, so that the wrist-plate, which is connected to the valves, can be swung on its pin. Now turn the eccentric around on the shaft

in order to determine the extreme points of travel of the wrist-plate. If all parts have been correctly adjusted, the line b will coincide with the lines n, n, at the extreme points of travel; if this is not the case, the hook rod will have to be adjusted at its stub end so as to obtain the desired equalized motion of the wrist-plate.

The next step will be to set the valves correctly with reference to the position of the crank; to do this the length of the rods K, K, L, and L must not be changed, but the following mode of procedure should be followed: Place the crank on one of its dead centers (see page 195) and turn the eccentric loosely on the shaft in the direction in which the engine is to run, until the steam valve nearest to the piston shows an opening or lead of $\frac{1}{32}$ to $\frac{1}{16}$ inch. After the proper lead has been given to this valve, secure the eccentric, and turn the shaft with eccentric in the same direction in which the engine is to run until the crank is on the opposite dead center, and notice if the opening or lead at this end of the cylinder is the same as on the other steam valve; if not, shorten or lengthen slightly, as may appear necessary, the connection between the wrist-plate and eccentric. Of course much adjustment in the length of these connections is not admissible without resetting the valves with reference to the wrist-plate. The compression on an engine is a very important factor, upon which cool and quiet running depends. With exhaust valves line and line about 5 per cent compression is secured, which is equal to $1\frac{3}{4}''$ for 36″ stroke and 2″ for 42″ stroke. In case more compression is desired, the exhaust valves must be given a little lap.

To set the exhaust valves for a given compression, say, 2 inches, first measure off 2 inches from the ends of the cross-head travel as shown in Fig. 138 (not from the ends of the guide). Then turn the crank in the direction it is to run until the end of the crosshead reaches the line on the guide. Adjust the exhaust valve corresponding to this end of the stroke so that it just closes

the port. Turn the crank over the center and back on the return stroke until the opposite end of the cross-head reaches the line on the opposite end (to the first mark) of the guide. Then adjust the exhaust valve corresponding to this end of the stroke so that it just closes the port. Both exhaust valves will then close the ports when the piston reaches a point 2 inches from the working end of the guide and the engine will then have exactly 2 inches

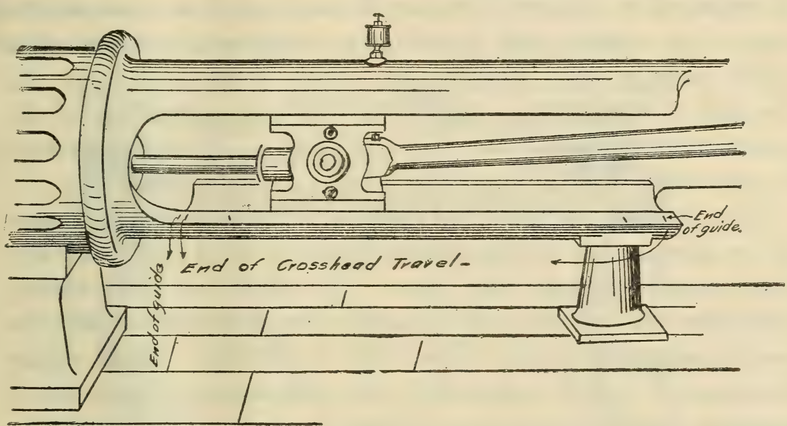


Fig. 138. Laying out compression marks on guide.

compression. If this is found to be too much or too little, as determined by the running qualities of the engine, it may be varied either way by adjusting the length of the rods L and L, being careful to turn each nut exactly the same amount.

The only thing which remains now to be done is to adjust the cam rods H, H, to produce an equal cut-off at each end of the cylinder. On the column of most Corliss engine governors will be found a stop device, sometimes in the form of a loose pin, some form of cam motion or movable collar. This device is for the purpose of preventing the governor from reaching its lowest position, for when it reaches the latter position the valves should not hook on. Should the governor belt break or become ineffect-

ive, the governor will stop and reach its lowest position on the column, thereby bringing the safety cam Y in underneath the inner member of the hook F which prevents the latter from engaging arm M, and as the valves cannot hook on when it is in this position the admission of steam to the cylinder is entirely shut off and the engine will come to a standstill.

It will be apparent that the stop on the governor column should be removed or otherwise rendered inoperative as soon as the engine has attained full speed, and should again be placed in active position when stopping the engine in the usual way. As the stop just mentioned determines the lowest position of the governor at which the valves should hook up, it should be kept in place while the foregoing adjustments are being made.

Next, unhook the reach rod from the wrist-plate and by means of the starting bar move the wrist-plate over until the lines b and n are nearly opposite each other. The head end valve should now have opened the port to nearly the limit, which may be ascertained by the marks on the ends of the valve. Now, adjust the governor rod H so that the projection or cam on the disk G operated by the governor will come in contact with the inner member of the steam hook F, so that the valve will be tripped or released when the marks b and n are exactly in line. As all governors do not move an equal amount to produce a given change in the point of cut-off, it will be safer to hook the reach rod on the wrist-plate and have the engine turned in the direction in which it is to run, until the head end valve is released, than to adjust the cut-off with the use of the starting bar only. To prove the correctness of the cut-off adjustment, raise the governor balls to a position where they probably would be when at work and block them there; then, with the connections made between the eccentric and the wrist-plate, turn the engine shaft slowly in the direction in which it is to run, and when the valve is released, measure upon the slide the distance which

the crosshead has moved from its extreme position. Continue to turn the shaft in the same direction, and, when the other valve is released, measure the distance through which the crosshead has moved from its extreme position, and if the cut-off is equalized, these two distances will be equal to each other. If they are not, adjust the length of the cam rods until the points of cut-off are equal distances from the beginning of the stroke. Replace the back bonnets and see that all connections have been properly made, which will complete the setting of the valves.

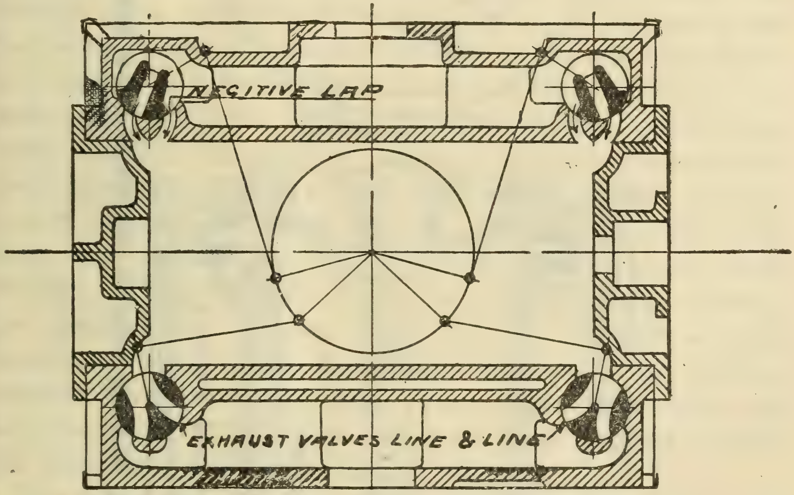


Fig. 139. Diagram of double eccentric gear.

ADJUSTMENT WITH TWO ECCENTRICS.

In order to obtain a greater range of cut-off in Corliss engines a separate steam and exhaust eccentric is used. With two eccentrics the admission and exhaust valves can be adjusted independently, and steam may be cut off anywhere, nearly to the end of the stroke.

The work of setting the valves of a Corliss engine having two

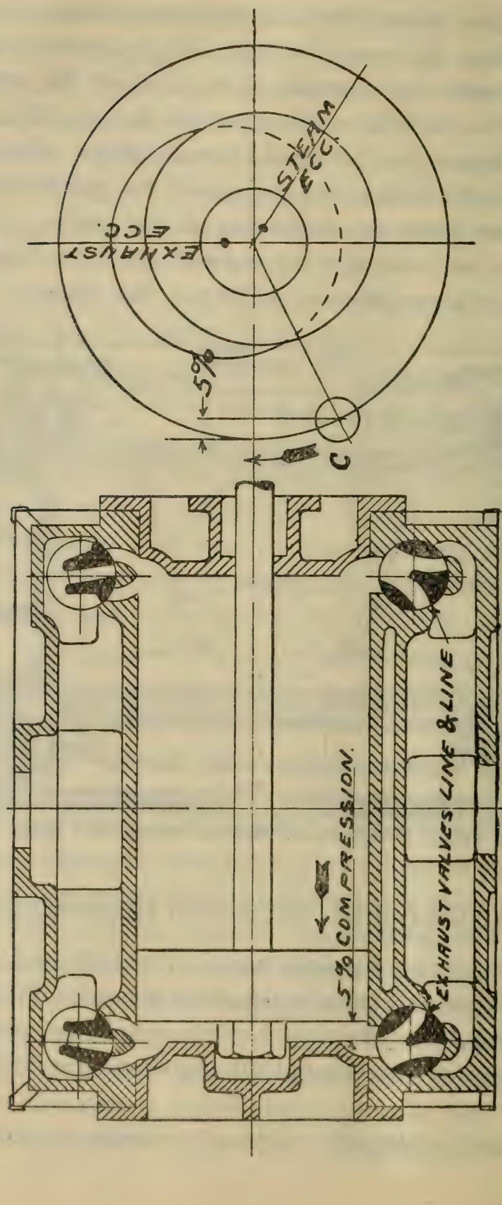


Fig. 140. Showing relative position of valves and eccentrics.

eccentrics is not particularly complicated as many engineers seem to think. After inspecting the type of releasing gear employed and knowing in which direction the engine is to run, finding the direction in which to turn the eccentric becomes a very simple matter. When setting the steam valves we have one eccentric to turn as in the case of the single eccentric engine, and when setting the exhaust valves another eccentric must be turned, but this does not add complication to the work, although it requires a little more time. The work of centralizing the positions of the various parts, equalizing the movements and setting and adjusting the valve gear is practically the same as with the single eccentric engine. Set the wrist-plate central as shown in Fig. 139, and adjust the valve rods; but in this case the steam valves are set with negative lap, which is usually a little less than half the port opening. The first step is to set the exhaust eccentric (as it is generally placed next to the bearing). To do this turn the engine until the piston is in the position shown in Fig. 140, so as to obtain a compression of about 5 per cent of the stroke. Then turn the exhaust eccentric loosely on the shaft in the direction the engine is to run, until the exhaust valves are line and line. Then secure the eccentric and turn the engine on the other end in the same position to prove the correctness of the other exhaust valve.

The next step is to set the steam eccentric; place the crank on either one of its dead centers, then turn the steam eccentric loosely on the shaft until the steam valve on the same end the piston is, has the required opening or lead, which varies from $\frac{1}{32}''$ to $\frac{1}{16}''$.

These directions apply to engines in which the reach rod from the eccentric is connected to the wrist-plate above the center pin R, Fig. No. 136. When the reach rod is connected to wrist-plate below the pin R, the eccentric should be turned the opposite direction to that in which the engine is to run.

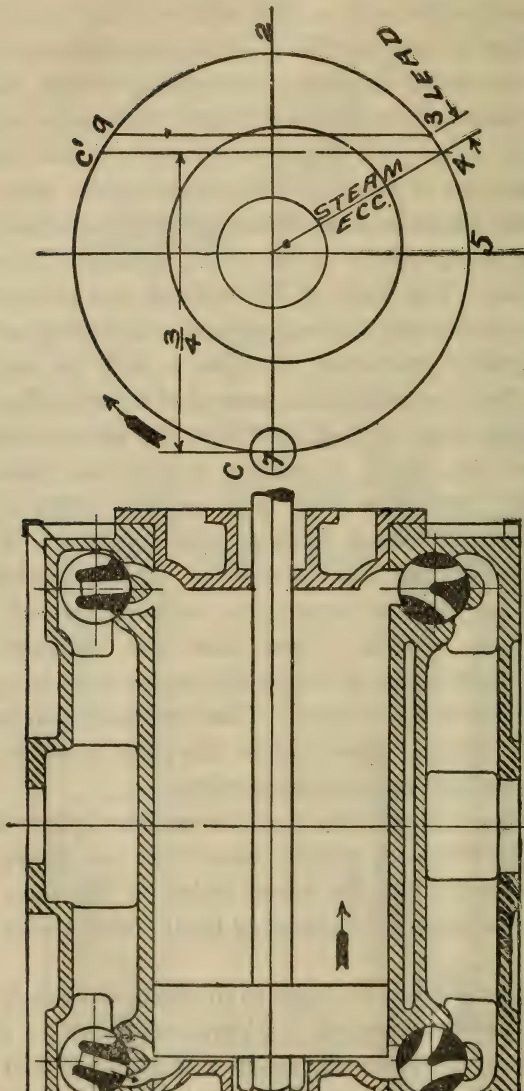


Fig. 141. Showing position of crank and eccentric at commencement of stroke.

The arrangement of the steam rods in Fig. 136 is in every respect satisfactory in connection with a single eccentric valve gear, for in that case a slow initial valve motion is imperative, and it is obtained by the lateral movement of the radius rod. But with two eccentrics quicker initial motion is feasible and desirable, and it is obtained by reversing the valve motion as in Fig. 139. Separate eccentrics require separate wrist-plates, which are usually placed on the same pin.

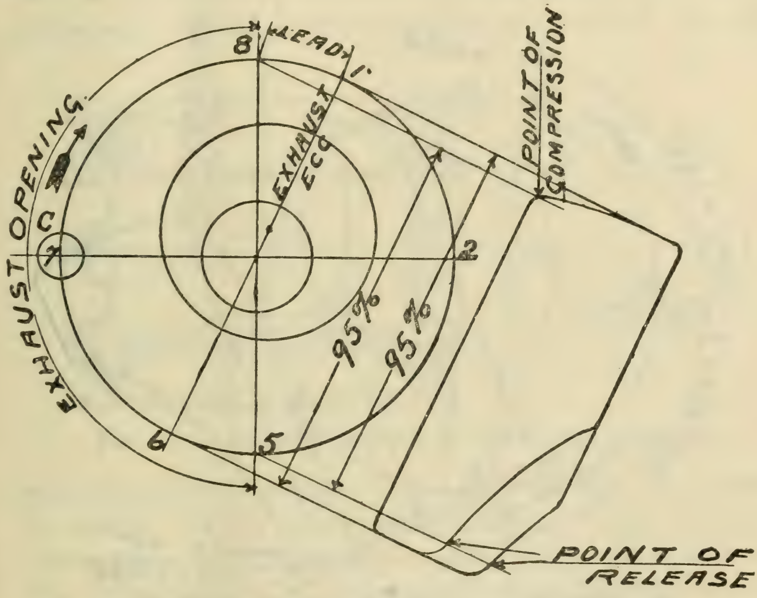


Fig. 142. Diagram showing steam distribution.

Figs. 141 and 142 show how the eccentrics may be placed on the shaft. The steam eccentric is at point 4, Fig. 141, the exhaust eccentric is at point 1, Fig. 142, and the crank is at its dead center at *C*. Individual eccentric circles are shown for the sake of clearness. An imaginary motion of the eccentric will point out the various events. Referring to Fig. 141, near point 2, at the end of

the throw, the hook connects with the steam valve; at point 3 the steam edges are at the point of separating and the eccentric motion 2–3 determines the initial valve motion. When the eccentric is at point 4 the crank is at its dead center as shown. At point 5 the steam wrist-plate is in its central position and in that position the valve does not cover the port, as with the single eccentric gear, but the port is open to a certain extent, determined by the eccentric motion 3–5. Point 7 marks the end

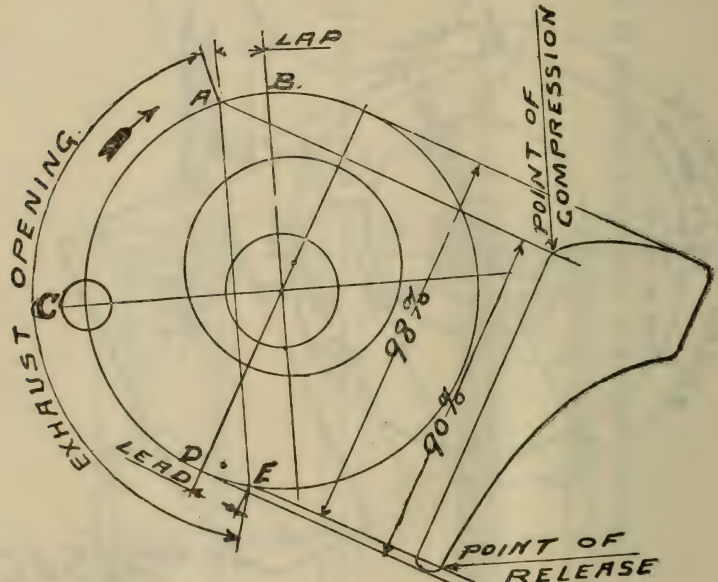


Fig. 143. Diagram showing steam distribution.

of the throw, and the corresponding position of the crank is at C^1 at about three-quarters of the piston stroke, and the limit of cut-off is a little later. If the hook does not strike the knock-off cam the valve will remain open until closed by the return stroke of the eccentric at point 9, near the middle of the return piston stroke. The exhaust action is discernible, Fig. 141.

It is similar to the single eccentric action, but with this difference, that the release at point 5 occurs at about 95 per cent of the stroke, and the exhaust is also cut off at about 95 per cent of the return stroke at point 8.

The motion of the exhaust valve after it has closed the port is determined by the eccentric motion 8–2–5, and full period of exhaust opening is obtained by the eccentric motion 5–7–8. In case the exhaust valve motion is designed and set with lap, Fig. 143 shows the effect lap has on the exhaust valves. The lap when wrist-plate is central is determined by motion *A-B*. It will be

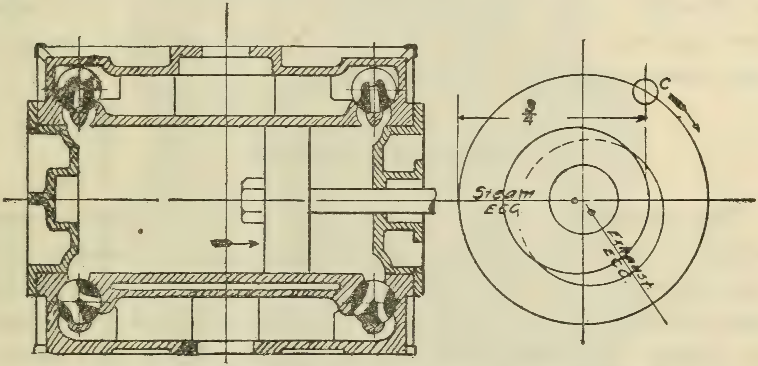


Fig. 144. Relative position of valves and crank pin.

noticed that the compression begins at *A* at about 90 per cent of the stroke, and the release at *E* occurs at 98 per cent of the return stroke and the exhaust opening *E*, *C*, *A*, is shortened. Where lap is used on the exhaust valve it has the effect of making earlier compression and later release. A valve gear designed to be operated by a single eccentric cannot very well be made to cut off much later than at half stroke, even when a separate exhaust eccentric is added. For the slow initial valve motion requires at least half the throw of the eccentric, and the other half is not sufficient for a late cut-off, and it will readily be seen from an inspection of Figs. 137 and 139, that a quicker initial valve motion in

Fig. 137 would involve radical changes in the valve gear. However, the range of cut-off may be extended by moving the eccentric back, sacrificing the lead, and to this there is no objections when it does not involve later release. The advantage gained by a second eccentric would consist in more compression and earlier release. After setting the valves and making the final adjustment, if it is convenient an indicator should be applied to the engine when at work to verify the adjustment of the valves for the best possible conditions for economical operation.

Fig. 143 indicates position of eccentric at $\frac{3}{4}$ cut-off, which can be extended some by giving the steam valves a little more negative lap, but as this shortens the amount of lap when closed, it may cause leakage in the steam valves.

COMPOUND ENGINE.

The compound engine is practically two single engines connected together and so arranged that the exhaust steam from one engine passes into and becomes the "live" steam for the other, in other words the first, or high pressure cylinder receives its supply of steam from the boiler and the second or low pressure cylinder receives its supply from the high pressure cylinder. The object of the compound engine is to enable the steam to expand to the lowest possible pressure with the least loss by condensation. When steam expands its temperature decreases, so that by the time the piston reaches the end of the stroke the temperature of the steam and consequently the temperature of the cylinder walls is considerably below the temperature of the incoming steam. The fresh steam of high temperature coming from the boiler comes in contact with the walls of the cylinder which have been cooled to the temperature of the exhaust steam, and the result is a considerable portion of the fresh steam is condensed, the latent heat serving to reheat the

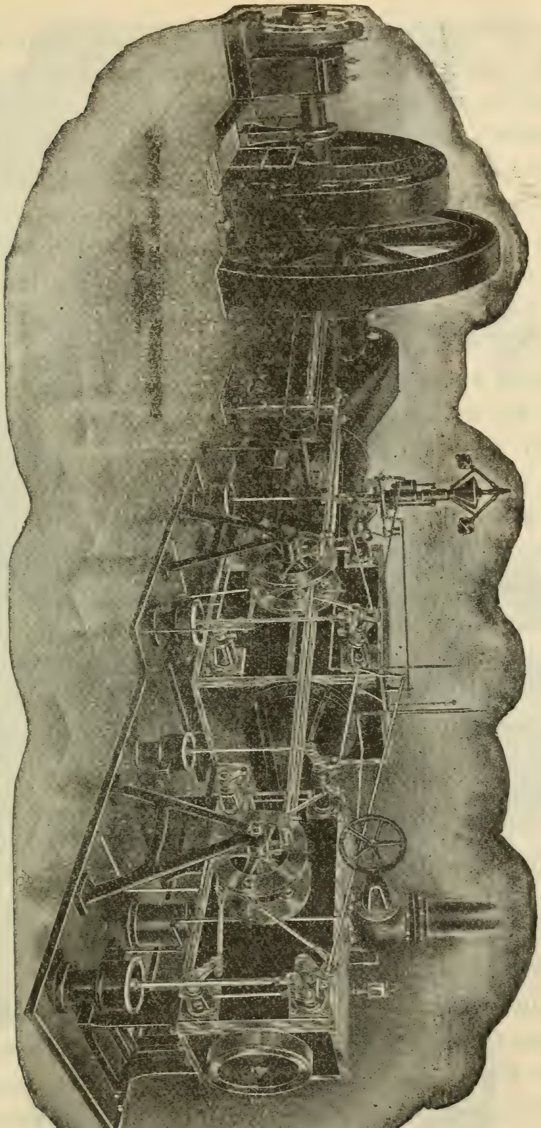


Fig. 145. Tandem compound Corliss engine.

cylinder walls. It will be understood that were it possible to keep the cylinder at a higher temperature, less steam would be condensed in warming it at each stroke and consequently more steam would be available for useful work. In the compound engine the steam is expanded partly in one cylinder and partly in the other so that the difference between the temperatures of the incoming and exhaust steam in each cylinder is greatly reduced. By this means steam may be expanded from a given initial pressure to a given final pressure with a loss of nearly twenty-five per cent less than would be incurred were the same expansion to take place in a single cylinder. It is due principally to avoiding the loss by cylinder condensation that the compound engine, considered as a type of engine, can perform nearly twenty-five per cent more work with the same weight of steam than can be obtained when the steam is expanded in one cylinder only.

In order to utilize the low pressure steam escaping from the high pressure cylinder it is necessary to provide a larger area of piston so that the low pressure steam acting on a large surface will do as much work as the high pressure steam acting on a smaller area. It is for this reason that the low pressure cylinder of compound engines is always made larger than the high pressure cylinder. The required size of low pressure cylinder for a given size of high pressure, depends upon the number of times the steam is to be expanded, the initial steam pressure and the nature of the work the engine is intended for. For steady loads the difference in the size of the two cylinders may be greater than where the load is constantly changing between wide limits as nearly always occurs in street railway service.

Compound engines, as this term is generally employed, are built of two types, the tandem compound, Fig. 145, and the cross compound, Fig. 146. In the tandem compound the work of both pistons is transmitted to the crank through one piston rod, cross-head and connecting rod, while in the cross compound there are

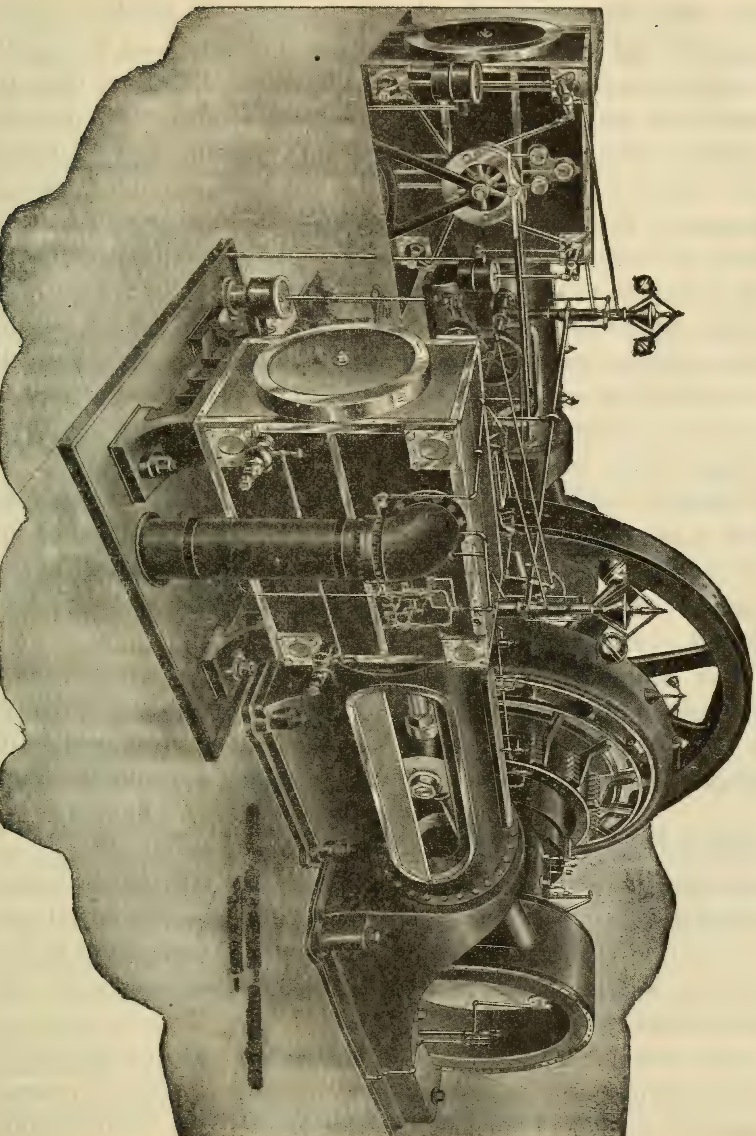


Fig. 146. Cross compound Corliss engine.

two complete engines placed side by side, the cranks of which are generally set 90 degrees apart. It will be seen that in the tandem compound engine it makes but little difference from the mechanical standpoint whether the work is divided evenly between the two cylinders or not because both pistons move in unison and drive the same crank. In the cross compound engine it is necessary, in order to secure a uniform turning effort at the shaft, to have the work divided as nearly equally between the two cylinders as the conditions will permit. In the tandem compound engine the principal consideration is the proper working of the steam, and the sizes of the cylinders are determined by the number of expansions to be effected in both cylinders, or the total number of expansions, as it is called, and the initial pressure. As the equal division of the work between the two cylinders in compound engines is essential, the ratio of the cylinders is generally for noncondensing $2\frac{1}{2}$ to 1 for 100 lbs., $2\frac{3}{4}$ to 1 for 125 lbs., and 3 to 1 for 150 lbs. initial pressure, and for condensing 3 to 1 for 100 lbs., $3\frac{1}{2}$ to 1 for 125 lbs., and 4 to 1 for 150 lbs., initial pressure.

The number of expansions required in a compound engine is represented by the quotient of the absolute initial pressure divided by the absolute terminal pressure. If steam is to be used at 105 pounds gauge pressure and is to be expanded down to 10 pounds absolute in the low pressure cylinder, there will be $\frac{105 + 15}{10} = 12$ expansions. A simple rule for finding the ratio of the area of cylinders for noncondensing, is to divide the absolute initial pressure by the terminal pressure which equals the expansions in both cylinders and the square root of total expansions equals the ratio of cylinders.

For example: 150 lbs. initial pressure plus 15 lbs. equals 165 lbs. absolute initial pressure, divided by 16 lbs. terminal pressure equals 10.3 total expansions, and the square root of 10.3 equals

3.2, equals ratio of cylinders. Care should be taken in non-condensing engines so that the ratio of the low pressure cylinder is not too large, as in such cases the steam in low pressure cylinder would expand to less than the atmospheric pressure, and thus make loops on indicator card which would incur a serious loss.

The calculation of the diameters of cylinders for a compound condensing engine when the data are given, follows. Take an engine that is to develop 500 horse power with an initial pressure of 105 pounds gauge, or 120 pounds absolute, the steam to be expanded to a terminal pressure of 6 pounds absolute. The total expansion of steam in both cylinders is $120 \div 6 = 20$.

Expansion in each cylinder $= \sqrt{20} = 4.47$.

Point of cut-off in each cylinder, per cent of stroke $= \frac{100}{4.47} =$ 22.3 per cent, 1 + hyp. log. of expansion in each cylinder = 1 + hyp. log. 4.47 = 2.497.

Terminal and back pressure in high pressure cylinder, and the initial pressure in the low $= \frac{120}{4.47} = 26.8$ pounds.

Mean effective pressure in h. p. cyl. $= 26.8 \times 2.497 - 26.8 = 40.11$ pounds.

Mean effective pressure in l. p. cyl. (assuming 3 lbs. back press.) $= 6 \times 2.497 - 3 = 11.98$ pounds.

If half the work is to be done in each cylinder, which is desirable in cross compound engines, each cylinder must do 250 horse power of work. Assuming the piston speed to be 600 feet per minute, the area of the low pressure cylinder is

$$\frac{33000 \times \text{H. P.}}{\text{Piston speed} \times \text{effective press.}} = \frac{33{,}000 \times 250}{600 \times 11.98} = 1147.7 \text{ square}$$

inches = 38 ins. diameter.

Area of high pressure cylinder by same rule is: $\frac{33{,}000 \times 250}{600 \times 40.11}$ = 342.3 square inches = 21 inches diameter.

$$\text{Ratio of cyl.} = \frac{40.11}{11.98} = 3.3 \text{ to one.}$$

The clearance and the areas of the piston rods have not been taken account of by separate processes in the foregoing calculations. These should always be included when making calculations involving the pressure and expansion of steam in engine cylinders. The method of finding the number of expansions taking place in a compound engine may be readily understood by referring to the diagram, Fig. 147. The shaded area in the

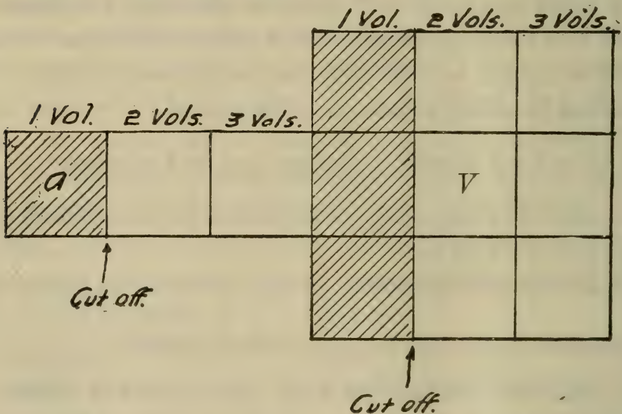


Fig. 147. Relative volume of high and low pressure cylinders.

smaller cylinder represents the initial volume of steam in the high pressure cylinder, that is to say, this represents the volume of steam taken from the boiler for one stroke, or during one-half revolution. The point of cut-off is at one-third stroke and the area of the low pressure cylinder is three times that of the high pressure cylinder. It will be seen that when the low pressure piston moves to one-third stroke the volume of the cylinder *V* behind the piston is equal to the volume of the entire high pressure cylinder. This shows that the capacity or contents of the low pressure cylinder is three times that of the high so that for every

volume of steam and therefore for every expansion taking place in the high pressure cylinder there will be three volumes, and three expansions taking place in the low pressure cylinder. This shows why the total number of expansions in a compound engine is the number in the high pressure cylinder multiplied by the number in the low pressure cylinder. In the diagram, Fig. 147, when the small piston reaches the end of the stroke the steam will have expanded three times, that is, it will occupy three times the space it did at the point of cut-off. Now when the large piston reaches the end of the stroke each of the three volumes a, a and a, Fig. 148, will have been expanded three more times and the total will be $3 \times 3 = 9$ expansions, that is, the original volume a, Fig. 147, will then occupy nine times the space it did when

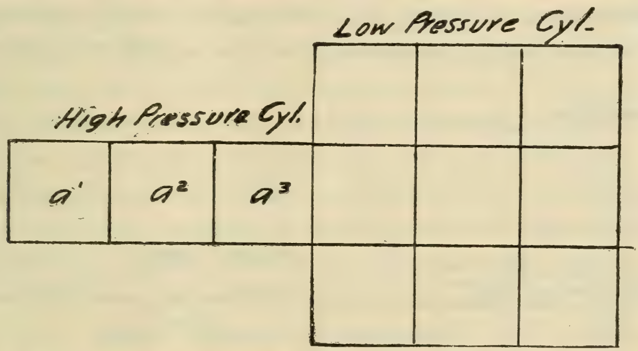
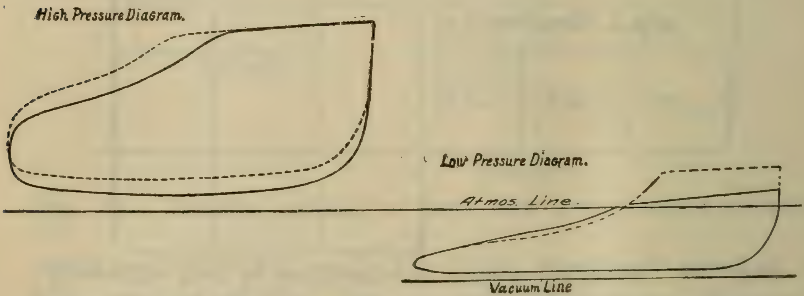


Fig. 148. Showing number of expansions in both cylinders.

first let into the high pressure cylinder. To find the number of expansions in a compound engine multiply the number of expansions in the high pressure cylinder by the number in the low, or multiply the number of expansions in the high pressure cylinder by the ratio of cylinder areas; the product will be the number required.

Again referring to Fig. 148, it will be seen that the low pressure cylinder must receive a high-pressure cylinderful of steam at each

stroke otherwise the pressure in the receiver and the back pressure on the high pressure piston will rise too high and a loss of power will result, or if the pressure be too low in the larger cylinder the small piston will drive the larger one which will again result in loss of power. It has been shown that the volume of both cylinders vary in proportion to the areas, that is, if the areas are as 1 to 3 then when both pistons have reached, say, one-third stroke the volume of one will be 3 times the volume of the other, and when the larger piston in this case travels one-third of the stroke the capacity of the low pressure cylinder behind the piston will then be equal to the whole of the smaller cylinder and will be capable of containing all the steam used during a full stroke of the smaller piston, or a high-pressure cylinderful of steam. This steam then expands during the remaining two-thirds of the stroke. Now it will be readily understood that if a cut-off valve were pro-



Figs. 149 and 150. Diagram from h. p. and l. p. cylinders.

vided on the low pressure cylinder and is set to cut off at less than one-third stroke (with a ratio of cylinder areas 1 to 3) the low pressure cylinder will not take a high pressure cylinderful of steam when steam is cut off, and the pressure in the receiver must necessarily rise. Reducing the volume of steam entering the low pressure cylinder apparently tends to lessen the work done by the larger piston and consequently more work must apparently be

done by the high pressure piston. This in turn causes a later cut-off in the small cylinder as shown in Fig. 149, dotted lines, which serves to neutralize the effect of the higher back pressure so that while the cut-off has been made later, the mean effective pressure remains practically the same. The higher back pressure on the small piston means a higher initial pressure in the low pressure cylinder, see Fig. 150 dotted lines, which causes more power to be developed in the latter cylinder. Thus it is seen that, within certain limits, shortening the cut-off in the low pressure cylinder puts more of the load upon the low pressure piston.

On the other hand when the low pressure piston is doing more work than the high pressure, the cut-off in the low pressure cylinder may be lengthened. This permits the low pressure cylinder taking more steam and consequently the receiver pressure and the back pressure on the high pressure piston are reduced and the work done by the high pressure piston is thus increased. By manipulating the cut-off on the low pressure cylinder the load on the two pistons may be equalized or very nearly so except when the engine is considerably underloaded or overloaded. The range of maximum economy is not as great with the compound as with the simple engine, that is to say, the load may be varied more widely from the point where the best economy is obtained, in the simple engine than in the compound which is due to the large difference in cylinder areas in the latter engine. At very early cut-off both the high pressure and the low pressure cylinders work the steam very similarly to the simple engine and as the loss by cylinder condensation increases with an increase in the range of temperatures it follows that an underloaded compound engine is but little if any more economical than a simple engine working with a similar initial point of cut-off.

In compound automatic cut-off engines the point of cut-off will be nominally the same in both cylinders, we say nominally (in name only) because the initial pressure and the extent of the

vacuum have some influence upon the receiver pressure and the mean effective pressure in the low pressure cylinder. In most compound engines in which the cut-off mechanism of both cylinders are operated by a single governor, provision is made for adjusting the cut-off of the low pressure cylinder relative to that in the high, so that while the nominal cut-off may be, say, one-fourth stroke, the actual points of cut-off may be one-fourth in the high pressure and $\frac{5}{16}$ in the low pressure cylinder, the governor, however, varying both points of cut-off as the load changes.

HORSE POWER OF COMPOUND ENGINE.

Little can be done in finding the horse power of compound engines without the indicator because of the uncertainty of the points of cut-off and consequently of the back pressure and mean effective pressures. The mean effective pressure in each cylinder may be computed by using assumed data, by the same rules given for simple engines, but it will readily be understood that assumed data furnishes assumed results only. Knowing the mean effective pressure areas and speed of the pistons the horse power of a compound engine is found as follows: Multiply the area of the high pressure piston by its mean effective pressure and divide by the area of the low pressure piston, then add this quotient to the mean effective pressure in the low pressure cylinder.* Call this answer 1. Multiply the area of the low pressure piston by the piston speed in feet per minute and by answer 1, and divide the last product by 33,000; the quotient will be the indicated horse power.

CONDENSING ENGINES.

It has been explained that the atmosphere exerts a pressure of about 15 lbs. per square inch on all surfaces with which it

* This quantity is to be taken as the M. E. P. when finding steam consumption of compound engine.

is in contact. The atmosphere is in contact with one side of an engine piston when the exhaust is open, and, consequently, the steam in pushing the piston forward, has to overcome this atmospheric pressure of 15 lbs. per square inch. The useful pressure of steam is, therefore, whatever pressure there is above the pressure of the atmosphere, and this is the pressure that the steam gauge shows. When the gauge says 60 lbs. we really have 75 lbs., but 15 lbs. of it does not count, because it is balanced by the atmospheric pressure on the other side of the piston. If we had sixty-pound steam pressing on the piston and could get rid of the atmospheric pressure on the side of the piston, the steam would exert a force of 75 lbs. per square inch, a very respectable gain, indeed. We might remove the air pressure by pumping it out, but the amount of power required in doing the pumping would be equal precisely to all gain hoped for, plus the friction of the pump; therefore, there would be an actual loss in the operation. But there is another way of removing the air pressure. It has been explained that a cubic inch of water vaporizes and expands into a cubic foot of steam at atmospheric pressure. If, after getting this cubic foot of steam, we take the heat out of it, we again turn it into the cubic inch of water. Assume the engine cylinder to hold just a cubic foot of steam, and assume that the stroke is complete and ready for the exhaust valve to open and permit this foot of steam to escape, and assume that this cubic foot of steam has expanded down to atmospheric pressure, that is, 15 lbs., absolute pressure. Now, instead of opening the cylinder to the atmosphere, we dose the cylinder with cold water. The heat leaves the steam and goes into the water and the steam turns to water, leaving in the cylinder the condensed steam in the form of a cubic inch of water. The steam formerly filled the cylinder, and now it fills but a cubic inch of it, consequently, we have produced in the cylinder a vacuum, which has the effect of adding about 15 lbs.

per square inch, to the force of the steam on the other side of the piston, by virtue of removing that much resistance to its forward motion. The heat, which was in the steam, has gone into the condensing water, except the trifle that remains in the cubic inch of condensed steam. We must get this condensed steam out of the cylinder, and it will be an advantage to pump it back into the boiler, for it is pure and it is hot.

This is the general principle of the condensing engine. It gives us the grand advantage of a heavy increase in the useful pressure acting to push the piston forward; it gives us pure water for use in the boiler, and it saves in the feed-water the heat that would otherwise go out of the exhaust pipe. But it is not practicable to condense the steam in the cylinder by dosing the cylinder with cold water. In practice, the steam is allowed to go into a separate condensing vessel, called the condenser. The condenser is precisely the opposite of the boiler. The boiler is the machine for putting heat into the steam to vaporize it, and the condenser is the machine for taking heat out of the steam and turning it into water again. In the condensing engine, one of these machines is pushing on the piston and the other machine is pulling on the piston. The gain by condensing is so great that it is a profitable piece of business to apply a condenser to any large non-condensing engine. The condenser requires a pump to withdraw the water of condensation, and this pump must be in reality an air-pump. In practice, they employ an air-pump and condenser combined in one structure, separate from the engine, and driven either by rod connection from the engine, or by a belt from the engine, or by an independent steam device. The arrangement will depend much upon the situation. The belt-driven pump permits of the condenser being set in any convenient position independent of the engine.

CONDENSERS.

When steam expands in the cylinder of a steam engine, its pressure gradually reduces and ultimately becomes so small that it cannot profitably be used for driving the piston. At this stage, a time has arrived when the attenuated vapor should be disposed of by some method, so as not to exert any back pressure or resistance to the return of the piston. If there were no atmospheric pressure, exhausting into the open air would effect the desired object. But, as there is in reality a pressure of about 14.7 pounds per square inch, due to the weight of the superincumbent atmosphere, it follows that steam in a non-condensing engine cannot economically be expanded below this pressure, and must eventually be exhausted against the atmosphere, which exerts a back pressure to that extent.

It is evident that if this back pressure be removed, the engine will not only be aided by the exhausting side of the piston being relieved of a resistance of 14.7 pounds per square inch, but moreover, as the exhaust or release of the steam from the engine cylinder will be against no pressure, the steam can be expanded in the cylinder quite, or nearly, to absolute 0 of pressure, and thus its full expansive power can be obtained.

Contact, in a closed vessel, with a spray of cold water, or with one side of a series of tubes, on the other side of which cold water is circulating, deprives the steam of nearly all its latent heat, and condenses it. In either case the act of condensation is

almost instantaneous. A change of state occurs and the vapor steam is reduced to water. As this water of condensation only occupies about one sixteen-hundredths of the space filled by the steam from which it is formed, it follows that the remainder of the space is void or vacant, and no pressure exists. Now, the expanded steam from the engine is conducted into this empty or vacuous space, and, as it meets with no resistance, the very limit of its usefulness is reached.

The vessel in which this condensation of steam takes place is the condensing chamber. The cold water that produces the condensation is the injection water; and the heated water, on leaving the condenser, is the discharge water. To make the action of the condensing apparatus continuous, the flow of the injection water and the removal of the discharge water, including the water from the liquefaction of the steam, must likewise be continuous.

The vacuum in the condenser is not quite perfect, because the cold injection water is heated by the steam and emits a vapor of a tension due to the temperature. When the temperature is 110 degrees Fahr.,. the tension or pressure of the vapor will be represented by about 4″ of mercury; that is, when the mercury in the ordinary barometer stands at 30″, a barometer with the space above the mercury communicating with the condenser, will stand at about 26″. The imperfection of vacuum is not wholly traceable to the vapor in the condenser, but also to the presence of air, a small quantity of which enters with the injection water and with the steam; the larger part, however, comes through air leaks and faultly connections and badly packed stuffing boxes. The air would gradually accumulate until it destroyed the vacuum, if provision were not made to constantly withdraw it, together with the heated water by means of a pump.

The amount of water required to thoroughly condense the steam from an engine is dependent upon two conditions: the total heat and volume of the steam, and the temperature of the injection

water. The former represents the work to be done, and the latter the value of the water by whose cooling agency the work of condensation of the steam is to be accomplished. Generally stated, with 26″ vacuum, the injection water at ordinary temperature, not exceeding 70° Fahr., from 20 to 30 times the quantity of water evaporated in the boilers will be required for the complete liquefaction of the exhaust steam. The efficiency of the injection water decreases very rapidly as its temperature increases, and at 80° and 90° Fahr., very much larger quantities are to be employed. Under the conditions of common temperature of water and a vacuum of 26″ of mercury, the injection water necessary per H. P. developed by the engine, will be from $1\frac{1}{4}$ gallons per minute when the steam admission is for one-fourth of the stroke, up to two gallons per minute, when the steam is carried three-fourths of the stroke of the engine.

WEIGHT OF WATER REQUIRED TO CONDENSE 1 POUND OF STEAM.

Temp. of Hot Well.	Back Press. in Cylinder. Lbs.	Temperature of Injection Water, Degs. F.					
		40	50	60	70	80	90
100	.94	17.8	21.4	26.8	35.7	53.5	107.
110	1.27	15.1	17.7	21.2	26.5	35.3	53.
120	1.68	13.1	15.	17.5	21.	26.3	35.
130	2.21	11.6	13.	14.9	17.3	20.8	26.
140	2.88	10.3	11.4	12.9	14.7	17.2	20.6

For other temperatures use the following formulas: —

JET CONDENSER.

Weight of water per lb. steam condensed $= \dfrac{1170 - T}{T - t}$

SURFACE CONDENSER.

Weight of water per lb. steam condensed $= \dfrac{H + 32 - T_1}{t_1 - t_2}$

In which T = temperature of hot well, t = temperature of the injection water, T_1 = temp. of condensed steam, t_1 = temp of circulating water leaving, and t_2, the temp. of the circulating water entering the condenser, H = total heat in steam above 32° F.

CHANGING FROM NONCONDENSING TO CONDENSING.

When it becomes necessary or desirable to change a simple noncondensing engine to a simple condensing engine, or to change a compound noncondensing to a compound condensing engine, a slight change in the adjustment of the exhaust valves of the simple engine and of the exhaust valves on the low pressure cylinder of the compound will in nearly all cases be found necessary in order to preserve the running qualities of the engine. The cause for this change may be explained as follows: In the first place it should be borne in mind that the pressure of steam varies inversely (oppositely) as the volume, and that when considering the volume and pressure of steam, all pressures are taken or measured above a perfect vacuum or above zero, in other words, only absolute pressures are to be considered. If 10 cubic feet of steam at zero gauge pressure, or at atmospheric pressure, which is 15 pounds absolute, be compressed into a space containing 5

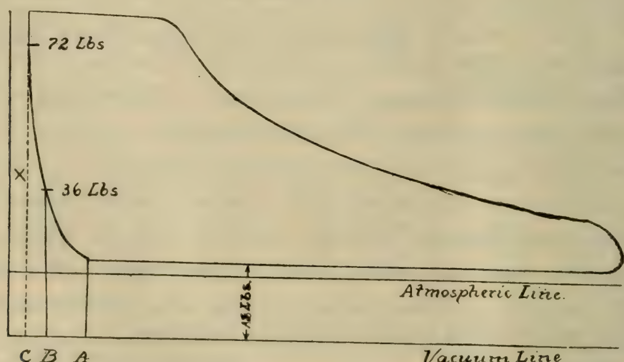


Fig. 151. Showing point of exhaust closure non-condensing.

cubic feet, the pressure of the steam will be raised to 30 pounds absolute, or 30 — 15 = 15 pounds by the gauge. Thus reducing the volume one-half serves to double the pressure. It will be

understood that this rule works both ways, namely, if the volume be doubled the pressure will be reduced one-half. Applying this principle to an engine, suppose the pressure during the return stroke in Figure 151 is 18 pounds absolute, and that the exhaust valve closes when the piston reaches the point *A*. The volume of steam entrapped in the cylinder at this point in the stroke is taken as the original volume. When the piston moves to the point *B*, the volume will have been reduced one-half, and consequently the pressure will have been doubled and will have risen to 36 pounds. When the piston reaches the point *C*, the volume will be but one-fourth of the original volume, and consequently the pressure will be four times the pressure of the original volume, or 18x4 = 72 pounds absolute, or 72 — 15 = 57 pounds by the gauge. As the space *x* represents the clearance, the steam cannot be further compressed so that the pressure of compression in this case is 72 pounds absolute, or 57 pounds gauge pressure. Suppose that the engine runs perfectly smooth and noiselessly with that amount of compression. If the compression be increased or decreased the engine will be apt to pound and possibly to run warm. Now suppose the engine be changed to condensing and that the back pressure be reduced to 4 pounds absolute. If the adjustment of the exhaust valves be not changed compression will begin at the same point in the return stroke. Referring to Figure 152, it will be seen by the full lines that the pressure of the original volume is now 4 pounds absolute. When the piston reaches the point *B*, the volume will be one-half of the original volume and consequently the pressure will be doubled, and will now be 8 pounds absolute. When the piston reaches the point *C*, the volume of steam will be reduced to one-fourth the original volume, and the pressure will be four times the pressure of the original volume, or 16 pounds absolute, which is 1 pound by the gauge. It will thus be seen that if the engine requires 57 pounds compression in order to run smoothly it cannot be expected to run smoothly with

only 1 pound compression by the gauge. In order to obtain the same pressure at the end of compression with steam of lower pressure the exhaust valve must close the port earlier in the return stroke. With the Corliss engine this may be accomplished by adjusting the radial rods connecting the exhaust valves with the wrist plate and in slide valve engines by adding exhaust or inside lap to the valve. To find the number of times the volume of the clearance x must be increased in order to lower the pressure from 72 pounds to 4 pounds absolute, divide the pressure of compres-

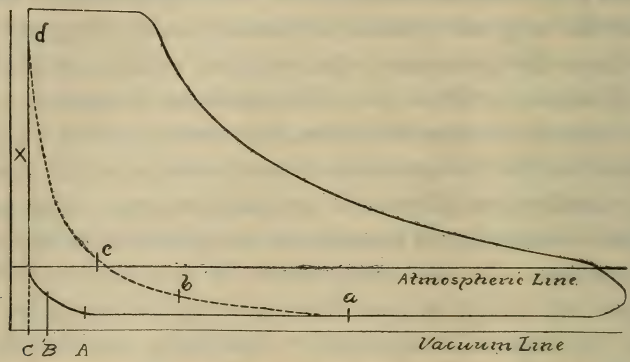


Fig. 152. Showing point of exhaust closure condensing.

sion by the back pressure thus $72 \div 4 = 18$ times. Now, by simply reversing this rule it will be seen that, in order that steam of 4 pounds may be compressed to a pressure of 72 pounds the volume must be reduced to $\frac{1}{18}$ of the original volume, therefore the original volume a must be equal to 18 times the volume of the clearance space x as shown by the dotted lines in Figure 152. Now, when the exhaust valve closes at a, Figure 152, the pressure is 4 pounds; at b the volume is but one-half, and the pressure 8 pounds; at c the volume is one-fourth and the pressure 16 pounds; at d the volume is $\frac{1}{18}$ of the original volume a, and the pressure is $18x4 = 72$ pounds, which is the pressure required for smooth

running. It will thus be seen that the same pressures are required whether running noncondensing or condensing, and that in order to get the same pressure of compression when running condensing as when running noncondensing, the exhaust valve must close earlier in the return stroke.

It will also be understood that when changing from condensing to noncondensing the order of things must be reversed, viz., the exhaust valve must close the port later in the return stroke because the higher the back pressure, the greater will be the pressure of compression. This explains why a condensing engine pounds and heats when the vacuum becomes impaired or is lost altogether. In this case the back pressure rises from that obtained when running condensing to the pressure obtained noncondensing and consequently the compression is much too high for smooth running. On the other hand when changing from noncondensing to condensing the back pressure is so low that scarcely any compression can be obtained, and if the engine requires considerable compression it is apt to pound and heat when the change is made provided the exhaust valves are not readjusted.

It frequently happens that the change from noncondensing to condensing is made at the end of the week, and in this case it is desirable to adjust the valves on Sunday to their approximate position by means of the marks on the valves, so as to avoid, as far as possible, any trouble when starting up on Monday morning. Under these circumstances it is desirable to estimate the extent of the change necessary in the adjustment of the valves. Multiply the length of the stroke in inches by the percentage of clearance and by one-fourth the absolute pressure of compression when running noncondensing; this product is the approximate point of exhaust closure when running condensing. Then lay off this distance, beginning at the ends of the guides. When the crosshead is within this distance of the end of

the return stroke, the exhaust valve should have just closed the port. For illustration, suppose an 18x42 Corliss engine is to be changed from noncondensing to condensing. The clearance is 3 per cent and the pressure of compression is 44 pounds absolute. The clearance is equal to $42 \times .03 = 1.26$ inches of the stroke, and one-fourth of the pressure of compression when running noncondensing is $44 \div 4 = 11$ pounds, so that the distance to be laid off on the guides is $1.26 \times 11 = 13.86$ or $13\frac{7}{8}$ inches, that is, when the back pressure is reduced to 4 pounds absolute by the condenser, the exhaust valve should close when the piston reaches a point $13\frac{7}{8}$ inches from the ends of the return stroke in order to obtain approximately the same compression as when running noncondensing.

It will be understood that this is merely a method of approximating the pressure and should not be relied upon except for making temporary adjustments.

The only reliable method of setting the exhaust valves when making changes of this kind is by means of the indicator because the pressures vary from several causes the effect of which cannot be calculated with a sufficient degree of accuracy for this purpose. This is true of both the simple and compound engines, the same method being used with the low pressure cylinder of the compound as with the simple or single cylinder engine.

TYPES OF CONDENSER.

Condensers may be divided into three general types or classes known as the surface, the jet and the siphon condensers. The latter type is sometimes referred to as the barometric condenser. The surface and jet condensers require an air pump and these two types are either direct driven or independent. The direct driven condenser is now little used for stationary work, being confined almost entirely to marine engines, and even there the independent

condenser is being very extensively used. The direct connected condenser is driven either by a belt or a direct connection to the crosshead of the main engine. The working vacuum is not obtainable in the direct connected condenser until after the engine is started and has made several revolutions. The disadvantage of the direct connected condenser is that the speed is always proportional to the speed of the engine so that the vacuum can only be regulated by changing the supply of injection or circulating water as the case may be. The independent condensers are driven by a steam cylinder forming a part of the condenser apparatus so that the vacuum can be maintained by regulating the supply of condensing water, and also by changing the speed of the air pump irrespective of the speed of the engine. The greater flexibility of the independent condenser enables it to maintain the working vacuum under widely changing loads and speeds. The cost of operating the latter type of condenser is greater than of the direct driven types because the power required for direct driving is obtained at the same cost per horsepower as that delivered by the engine, while the cost of operating the independent condenser is practically the same as that of an ordinary steam pump. This apparent loss can be largely avoided by utilizing the exhaust steam from the condenser for heating the feed water.

The following illustrations represent the three principal constructions of condenser to be found in the average stationary practice.

A sectional view of the surface condenser with air and circulating pumps attached is shown in Fig. 153. It consists of a shell, usually of cast iron, containing a large number of small brass or copper tubes through which the condensing, or circulating water as it is called, is pumped by the circulating pump. The water enters the chamber at one end of the condensing chamber and flows through one bank of tubes into a chamber at the opposite end and thence back again and out at the top into

the discharge pipe. The steam enters the condensing chamber at the top where the current is divided by a baffle plate, which sends the steam in both directions and distributes it more evenly over the cool tubes, the steam being condensed by coming in contact with the tubes, the temperature of which is the same as that of the circulating water flowing through them. The condensed steam collects in the bottom of the condensing chamber and flows into the suction pipe of the air pump, which also removes the air and aids in maintaining the vacuum. The steam cylinder

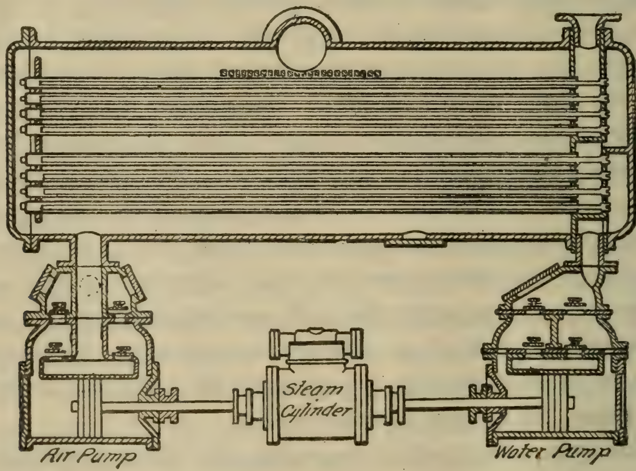


Fig. 153. Diagram of Surface Condenser.

is placed between the air and circulating pumps, the pistons being connected to the same piston rod.

Surface condensers embody two constructions or arrangements of tubes. In one the tubes are connected to tube plates, the arrangement being similar in every way to the return tubular boiler. The other construction is known as the double tube condenser in which two sets of tubes are employed, one within the other as shown in Fig. 153. The circulating water flows through

the inner tubes and returns through the outer or larger ones. This construction has the advantage in that the outer tubes are kept at practically the same temperature throughout their length, thus increasing the efficiency of the tube surface. Water will absorb heat more rapidly when flowing at a high velocity than at a low velocity, consequently the double tube condenser can be made smaller for a given capacity than the single tube type. The air pump discharges into the hot well, and as the circulating water does not come in contact with the steam, either before or after it is condensed, the water in the hot well is always pure, or nearly so, and suitable for use in the boilers. When the surface condenser is employed the steam passes from the low pressure cylinder into the hot well and consequently some provision should be made for removing the oil from the exhaust steam. Greasy condenser tubes are not as efficient as clean ones and for this reason it is advisable to place the oil extractor between the low pressure cylinder and the condenser. Any oil that may pass through the extractor and into the hot well may be avoided by taking the supply to the boiler feed pump at a point below the surface of the water in the hot well.

The surface condenser is particularly valuable where the water is unfit for use in the boilers and for this reason it is used more largely for marine than for stationary work.

The surface condenser furnishes the more reliable means of measuring the steam consumption of engines and pumps because the discharge of the air pump in a given time represents exactly the quantity of water required by the engine in the same length of time.

Fig. 154 is a sectional view of the independent jet condenser. The external appearance, and sometimes the minor details of construction, will vary slightly in this style of condenser but it is fortunate for engineers that the principles involved and the practical operation of jet condensers are precisely the same in all.

The steam and water cylinders and valves of the jet condenser are identical to similar parts of the ordinary direct acting pump, in fact, the jet condenser is a direct acting pump with a simple condensing chamber connected to the pump suction. The condensing chamber resembles an air chamber in form, except that it

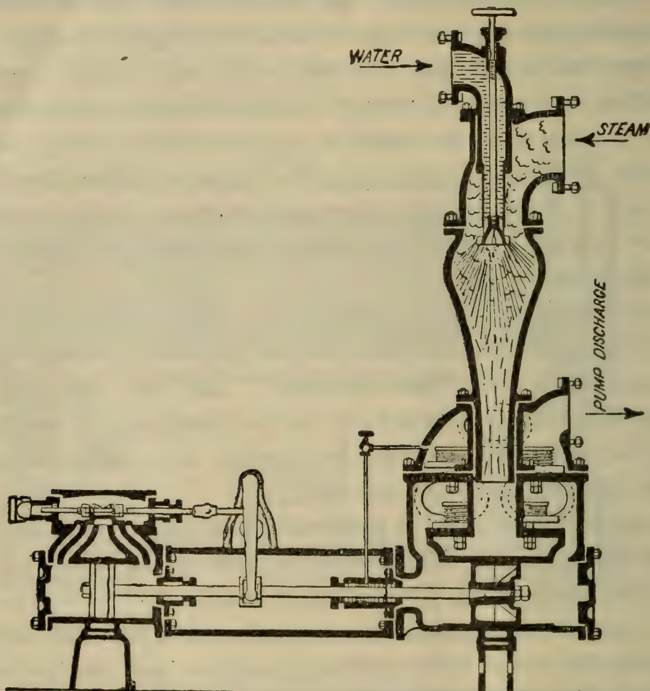


Fig. 154. Independent Jet Condenser.

is larger, and it is provided with an inlet for the exhaust steam and an outlet for the injection or condensing water. The inlet for the water is carried down to the spherical part of the condensing chamber and terminates in a cone-shaped spraying device, which throws the water out in an umbrella-shaped film against the sides of the chamber. The exhaust steam enters above the

spray and cannot reach the pump below without passing into the spray of cold water, which condenses it. The mixture of condensed steam and injection water is then drawn into the pump and is finally discharged into the hot well. The admixture of the condensed steam with the injection water improves the quality of the water in the hot well, provided means are employed for removing the oil from the exhaust steam, thus rendering the water better fitted for use in the boilers. The injection water is regulated by means of a hand wheel at the top of the condensing chamber which regulates the position of the cone at the end of the water inlet.

When engines are lightly loaded, or liable to be for any length of time, it is especially desirable to provide a safety device which

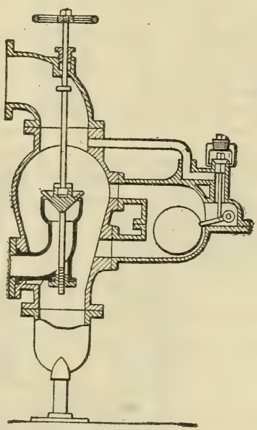


Fig. 155. Device for Breaking the Vacuum.

will break the vacuum when the water rises to a certain height in the condensing chamber. If this is not done injury to the engine is liable to occur at any time because should the speed of the pump slacken, thus permitting water to accumulate in the condensing chamber, the vacuum will be impaired and the low

pressure cylinder will then act as an air pump, owing to the small amount of steam admitted and the low pressure due to expansion, and will thus draw the water into the cylinder probably wrecking the cylinder completely. A simple safety device adapted to prevent injury to the engine as the result of flooding the condenser is illustrated in Fig. 155. It consists of a simple float con-

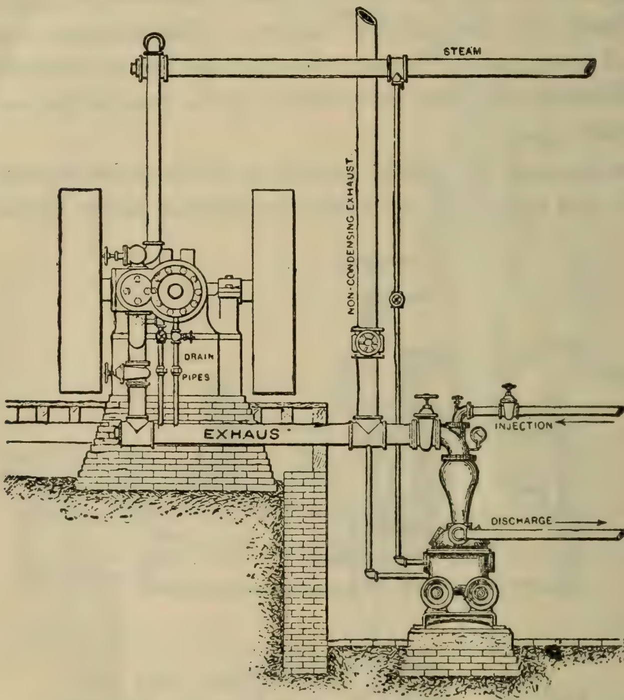


Fig. 156. Method of connecting up a Jet Condenser.

nected to an air valve at the side of the condensing chamber so that when the water level rises high enough to raise the float, air is admitted and the vacuum immediately destroyed. The exhaust steam from the engine then flows out through the atmospheric relief valve into the atmosphere, and the engine is con-

verted into a noncondensing engine automatically and without stopping.

Jet condensers will raise water for condensing purposes from 16 to 20 feet although it is desirable to keep the lift as low as possible because the higher the lift, the more it costs to operate the condenser and consequently the net saving by running condensing is correspondingly less.

The jet condenser requires precisely the same care as the direct acting pump except that greater pains must be taken to prevent the leakage of air and water. An engineer who can keep a direct acting pump in first-class running order need have little trouble in caring for a condenser whether it be large or small.

Fig. 156 illustrates the method of connecting the jet condenser with the engine. When the exhaust steam from the condenser cannot be profitably used for heating the feed water it can be turned into the engine exhaust pipe, or into the condensing chamber direct, and the condenser thus be made to run condensing also.

The siphon condenser is illustrated in Fig. 157, and is the simplest form of condenser in use at the present time. The exhaust steam enters the somewhat contracted condensing chamber through the cone-shaped nozzle, while the condensing water enters at the side and surrounds the cone, the water issuing into the chamber below through the annular orifice formed between the cone and the walls of the condensing chamber. The water issues at rather high velocity and expels the air from the exhaust pipe. The steam upon leaving the cone-shaped nozzle flows into an inverted cone, formed by the film of water issuing from the annular orifice which condenses the steam. The condensed steam and injection water flow at rather high velocity down through the contracted neck, where the stream is solidified, into the tail pipe, thence into the hot well. The neck of the condensing chamber is contracted, thus forming a combining tube, for the purpose of

giving the water sufficient velocity to maintain a siphon-like action that draws the steam from the exhaust pipe and causes a vacuum to exist in it. This style of condenser should be placed 34 feet above the level of the water in the hot well. An atmospheric discharge or relief valve is placed at the top which opens automatically when the vacuum is destroyed thus permitting the exhaust steam to flow into the atmosphere. The siphon

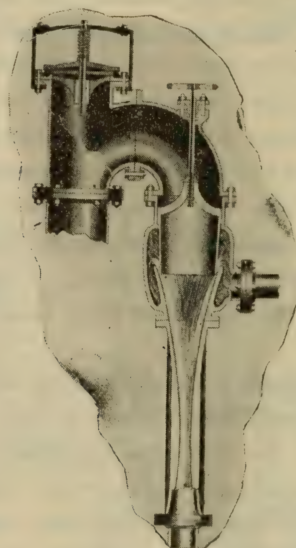


Fig. 157. Siphon Condenser.

condenser will raise the condensing water to a height of from 15 to 18 feet so that when the water supply is located within that distance of the top of the condenser no pump is required, the condenser continuing to siphon over the water as long as steam is condensed. When the condenser is operated without a pump a starting pipe will be necessary; this pipe connecting the water supply with the tail pipe as shown in Fig. 158. When starting the condenser, the valve in the starting pipe is opened, and the

water flowing through this and down the tail pipe will gradually exhaust the air from the upper part of the tail pipe until sufficient vacuum is formed to draw the water up to the condenser and start the water flowing through it. When this is done the valve in the starting pipe is closed, and the engine started.

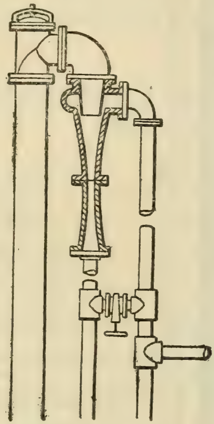


Fig. 158. Siphon Condenser and Starting Valve.

Fig. 159 shows the method of connecting a siphon condenser with the engine. In this illustration the condensing water is supplied by a pump.

It frequently becomes necessary to destroy the vacuum and to stop the engine on short notice, and for this reason it is desirable to have some means of opening the atmospheric relief valve at the top of the condenser. This is especially desirable when the condenser is arranged to siphon the water because any failure of the condenser with a light load on the engine would result in water being drawn into the low pressure cylinder and probably wrecking it. An arrangement for thus breaking the vacuum is shown in Fig. 159.

The water in the hot well consists of a mixture of condensing

water and condensed steam so that if the water is to be used in the boilers means should be provided for removing the oil from the exhaust steam, and the supply to the boiler feed pump

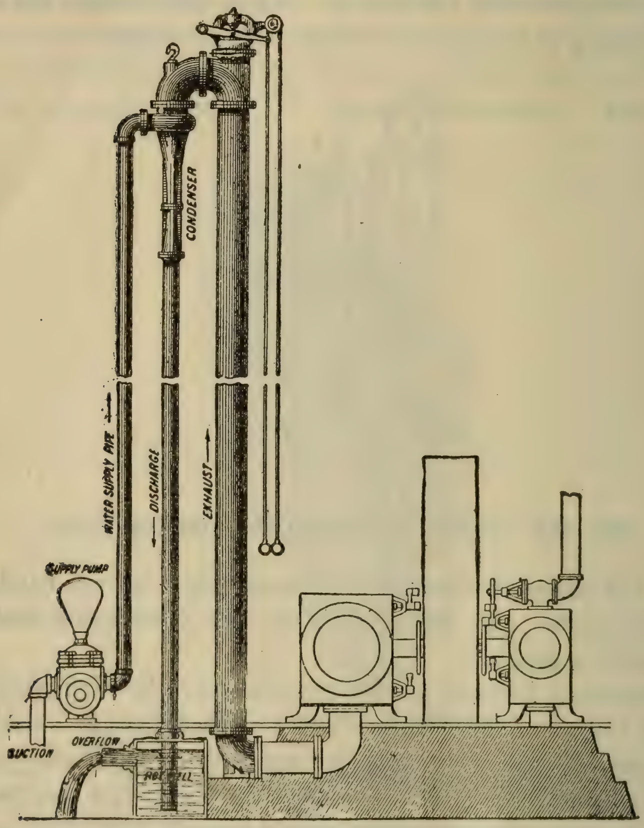


Fig. 159. Siphon Condenser connected to Engine and Pump.

should be taken at a point considerably below the water level in the hot well.

The siphon condenser, when properly proportioned and connected up maintains a good vacuum and possesses the advantage

that it contains no moving parts and nothing to get out of order, while it is adapted to engines of all sizes, and under all conditions which will permit it to be elevated to a height of 34 feet above the level of the water in the hot well.

The quantity of condensing water required by the siphon condenser is the same as for the jet condenser and may be found by means of the formulas on page 237.

STARTING AND RUNNING A COMPOUND CONDENSING ENGINE.

The principal aim of the engineer in charge of a plant is to keep his engine running, and to run safely, and when running condensing the object also is to maintain the vacuum. The vacuum is formed by removing the air from the exhaust pipe and its immediate connections, including the condenser. Therefore when air leaks in or is admitted through a valve either intentionally or otherwise the vacuum will at once be destroyed. It will be understood from this that all air leaks should be carefully stopped. The principal points to receive attention when testing for air leaks are, the stuffing boxes on both the engine and the condenser, and also on the valves in pipes leading to and from the condenser. A condensing engine and its condenser should be provided with a solid foundation and be securely bolted to it. Vibration in the piping is a common cause of leaky joints, and air leaks will affect the vacuum very seriously. It is a good plan to test the exhaust and condenser piping occasionally while the engine is running. A lighted candle held close to the joints will serve to locate a leak, which should be marked with a piece of chalk so as to be readily found when an opportunity occurs to make repairs.

The vacuum may be reduced and even destroyed by too much water as well as by too little. When too little water is admitted all the steam is not condensed, and the accumulation of uncon-

densed steam raises the pressure in the condenser until finally the back pressure valve opens and the engine exhausts into the atmosphere and, of course, becomes a noncondensing engine. When too much water is admitted the air pump becomes flooded, the volume of the condenser is decreased, the injection interfered with, and air gradually accumulates, causing the vacuum to become less, sometimes rapidly and at others quite gradually depending on the excess of water.

The condenser should be run at as low a speed as possible and be able to discharge the necessary quantity of water, and to maintain the required vacuum. When air leaks occur the speed of the condenser must be increased with the same load on the engine and with other conditions the same, so that when it becomes necessary to gradually increase the speed it indicates that leakage is occurring either at some point in the exhaust piping, in the injection pipe leading to the condenser or in the valves or piston in the condenser itself. It is not profitable to allow the packing in the stuffing boxes on the engine and condenser, and on the valves in the piping, to remain as long as it will apparently work air tight because when old packing begins to give out it generally becomes useless in a very short time, and, furthermore, it seldom admits of any adjustment with beneficial results. It is better to renew the packing at shorter intervals and know that when a gland is tightened the leakage will be stopped.

Generally speaking the higher the vacuum the better, but this is not always true. When engines are very lightly loaded and have but little resistance above that due to friction, it is sometimes better economy to reduce the vacuum, thus slightly increasing the period of admission, because the engine uses steam expansively and the condenser does not, so that a part of the steam required by the condenser can be used to better advantage in the engine cylinder. It sometimes occurs that the boilers are too small, and where an exhaust steam feed water heater is placed in the

low pressure exhaust pipe it will frequently be found more economical to reduce the vacuum and thus send the feed water to the boilers at a higher temperature. The regulation of the vacuum under ordinary conditions should be governed by the position of the governor as well as by the vacuum gauge, the object being to maintain a vacuum that will keep the governor at the highest position. Under certain conditions of load the cut-off will be found to be shorter with 26 inches than with 27 or 28 inches vacuum. The receiver pressure also has something to do with the position of the governor. The receiver is usually a plain metal cylinder placed beneath the engine room floor, and to which the exhaust pipe from the high pressure cylinder and the steam pipe to the low pressure cylinder are connected. The high pressure cylinder exhausts into the receiver and the low pressure cylinder takes its steam from the receiver. It will be understood from this that the receiver virtually serves as the boiler for the low pressure cylinder. The pressure in the receiver is generally a little lower than the terminal pressure in the high pressure cylinder, while the receiver pressure and the initial pressure in the low pressure cylinder are practically the same.

What is known as a reheater, which is sometimes used in connection with compound engines, is merely a coil of steam pipe placed in the receiver, steam of higher temperature than that of the exhaust from the high pressure cylinder being admitted to the coil which heats the steam in the receiver to a temperature higher than that due to the pressure, thus serving to reduce the loss due to condensation in the low pressure cylinder, and incidentally improving the economy of the engine as a whole. Changing the receiver pressure will oftentimes alter the position of the governor by improving the distribution of the load between the high and low pressure cylinders. It should be remembered in this connection that lengthening the point of cut-off in the low pressure cylinder reduces the receiver pressure and consequently tends to

throw more of the load on the high pressure cylinder, while shortening the cut-off in the low pressure cylinder tends to throw more of the load on to the low pressure cylinder. The receiver pressure is governed by two things, viz., the amount of steam put into it, and the amount of steam drawn out of it. If the low pressure cylinder draws more out of it than the higher pressure cylinder puts into it, the pressure must fall because the volume is thus increased. Engine cylinders are generally so proportioned that the receiver pressure will be from 4 to 6 pounds lower than the terminal pressure in the high pressure cylinder when the engine has a fair load. When an engineer knows what the terminal pressure in the high pressure cylinder is it is not difficult to set the cut-offs to produce the desired results. When the high pressure exhaust valves pound and lift from the seats it may be safely assumed that the receiver pressure is higher than need be, and it will generally be found practicable to lessen the pressure, which may be done by lengthening the period of admission in the low pressure cylinder, being careful to note exactly the extent of the adjustment because there is the possibility of other things causing the valves to pound, and in that case the cut-off should be shortened again exactly the same amount to which it had been lengthened.

For usual ratios of cylinder volumes, viz., from 3 to 1 or 4 to 1, and under average conditions of load on condensing engines, a boiler pressure of 80 or 90 pounds will produce a receiver pressure of about 5 pounds; 100 pounds will produce from 7 to 10 pounds; 125 pounds from 12 to 15, and 150 pounds from 18 to 20 pounds. The receiver pressure frequently rises from 1 to 3 pounds with changes of load.

The best results are generally obtained when the load is equally divided between the high and low pressure cylinders. The governor should thus remain at the highest obtainable point with any given load and with a moderately high vacuum, say 26 or 26½ inches. The extent of changes to produce these results cannot

be foretold, but must be ascertained by experiment in each case.

When about to start a compound condensing engine the first thing to be done is to see that all the water is drained out of the piping. If the engine is a Corliss or other four-valve engine the high pressure cylinder will readily get rid of the water, but this water drains into the receiver, and if not removed will enter the low pressure cylinder when the engine is started, and probably wreck it. If there is a separator it should be carefully drained.

Then the pipe above the engine throttle, then the cylinder and lastly the receiver. If these parts are thoroughly drained before starting, what little water may enter the low pressure cylinder, due to condensation in the high pressure cylinder and receiver, will cause no trouble or damage provided the engine is started slowly so as to give the water time to pass out through the exhaust ports. The steam line to the condenser should also be drained down to the condenser throttle so that when it is time to start the condenser it can be done without delay.

About 20 minutes before time for starting the plant begin to warm the cylinders. Place the wrist plates at their central position. Open the throttle a little and also the live steam valve to the receiver. Allow the receiver drain to remain open a little until after starting. Then open the oil cups and the cylinder lubricators and see that they are feeding the proper quantity of oil.

If it is not desirable for any reason to start the engine in advance of the other machinery, the wrist plates can be worked back and forth a few times by hand, after which the throttle and receiver valve should be closed, leaving the wrist plates in the central position. A little practice will indicate how wide the throttle can be opened and how high the receiver pressure may be allowed to rise without moving the engine when working the wrist plates. After warming the cylinders, the condenser may be

started. If the air pump is in proper running order the vacuum gauge will soon indicate the vacuum, which ought to be increased to about 25 inches. If the exhaust from the steam cylinder of the condenser is run into the condensing chamber, the injection water should be turned on after the air pump has made three or four strokes, otherwise the injection water may be turned on immediately upon starting the engine, or before if desired. When the running position of the injection valve is not known it may be tried at from one-half to two-thirds open. When the air pump is running properly inspect the overflow to see that the water is discharged properly and freely. Considerable water will be discharged from the condenser if the injection valve is open and the condenser is working properly.

Returning to the engine, hook in the low pressure reach rod and then the high pressure. Then open the throttle a little and start the engine slowly so that any slight condensation which may have accumulated can be worked out more slowly. After the engine has made one or two revolutions attention should be paid to the vacuum gauge to see that the condenser is getting the proper amount of water. It is probable the injection valve will have to be opened wider. The engine should be worked up to speed gradually so that proper attention can be given to the regulation of the injection water as the steam flowing into the condenser gradually increases.

When the engine is up to speed and before the load is thrown on it is a good plan to go to the end of the condenser discharge pipe and find out how hot the water is. This may be done with the hand. The water should be decidedly cooler than the hand, but if not, more injection water must enter the condenser. When the load is thrown on, it is a safe plan to again test the discharge water and if the condenser is working properly it will now be perceptibly warmer than the hand. It may be about as warm as a person would ordinarily heat water for washing the hands.

When the discharge becomes so hot as to be unbearable to the hand it is a sign of impending danger, viz., that of losing the vacuum.

It is a good plan to have a small auxiliary injection pipe connected to the city mains or to the delivery pipe of any cold water pump, which can be used to supplement the main injection in emergencies of this kind. When a jet condenser lifts its own water, it must lose the suction, as it is called, whenever it loses the vacuum, so that without the aid of the auxiliary injection pipe it can seldom regain the vacuum without allowing the engine to exhaust into the atmosphere until the condenser can be cooled. When the load suddenly increases and the discharge becomes too hot, the normal temperature can generally be quickly restored by opening the auxiliary injection valve. The auxiliary injection is generally provided on condensers. It may enter anywhere in the exhaust pipe. An auxiliary injection can be made of a piece of 1½ or 2-inch pipe about 3 feet long, perforated with small holes so as to make a sprayer head, the sprayer head being connected with the same sized pipe from the city main or pump. The auxiliary injection is in every sense an emergency apparatus and should be carefully watched while in operation, and shut off as soon as the need of it ceases because it is not automatic and if allowed to run unnoticed there is danger of flooding the condenser and exhaust pipe and of working water back to the engine. When the load on the engine is fairly steady, or when the load fluctuates uniformly between certain limits the condensing apparatus will require very little attention. The same precautions should be taken with a condenser that raises its own water as with a boiler feed pump, viz., to see that nothing interferes with its continued operation, that the suction pipe is properly protected so that no obstruction may enter and that the discharge is always free.

It is sometimes necessary to change from noncondensing to condensing without stopping the engine. In this case the gate

valve in the exhaust pipe leading to the condenser will be found closed because this valve should always be closed when the condenser is not in use, otherwise the steam would be apt to injure the valves in the condenser. The air pump is first started and the injection valve opened in the same manner as when about to start the engine, and is opened to the running position with a load. When the condenser is working properly an assistant should be stationed at the back pressure valve ready to close it when signaled by the engineer. The gate valve in the exhaust pipe is then partly opened, the engineer watching the vacuum gauge to note the point at which the vacuum begins to drop. At that moment he signals the assistant, who then closes the back pressure valve. The gate valve is then opened wide. When opening the gate valve, and before the back pressure valve is closed, care must be taken not to continue to open the valve after the vacuum begins to fall, otherwise the vacuum will be quickly lost entirely.

When changing from condensing to noncondensing all that is necessary is to close the gate valve in the exhaust pipe and stop the condenser. As soon as the pressure in the exhaust pipe equals that of the atmosphere, the back pressure valve opens automatically.

The surface condenser is worked in the same manner as the jet condenser, which has just been described. There is an extra pump, which must keep the circulating water flowing through the condenser tubes; that much work being taken off the air pump. The air pump has air and some water to handle in the same manner as though it pumped the cooling water. Both these pumps, together with the pump of the jet condenser, require the same care and attention that the boiler feed pump does or any pump that lifts water by suction.

When stopping a condensing engine, stop the engine first, then the condenser, then shut off the oil cups and lubricators and lastly open the drips.

SETTING THE PISTON TYPE OF VALVE.

The simple piston valve admitting steam between the pistons is, in operation, the reverse of the plain *D* slide valve, which admits steam at the outer edges, or ends of the valve. To make this still clearer it may be said that were the live steam to enter through the exhaust cavity of the *D* slide valve its operation and the position of the eccentric relative to the crank would be iden-

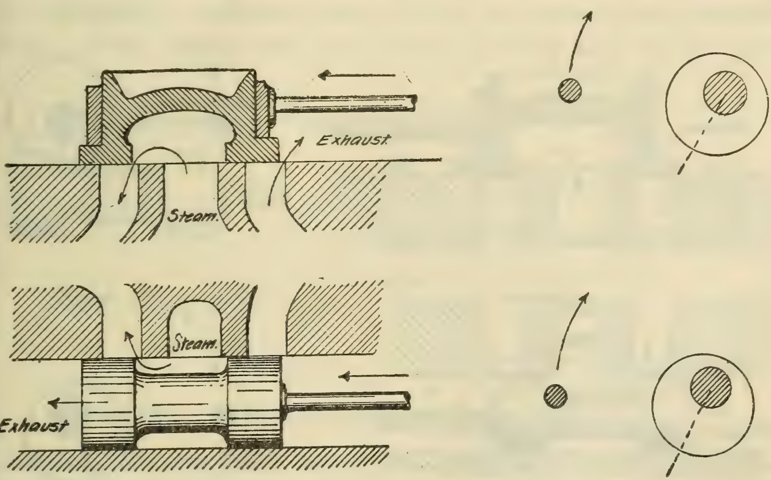


Fig. 160. Similarity between the slide and piston valves.

tical to the piston valve. Fig. 160 illustrates the similarity of action and eccentric positions were these conditions to obtain.

In these types of valve, as ordinarily employed, the steam is admitted at the ends of the slide valve, and between the pistons or at the middle of the piston valve. The change from the end to the middle of the valve necessitates a change in the position of the eccentric relative to the crank in order to have the direction of rotation remain the same. The positions of the eccentric when driving the simple *D* valve, and the piston valve, are indi-

cated in Fig. 161. It will be noticed that the crank revolves in the same direction in both cases, and that when the crank leaves the dead center, moving in the direction of the arrow, the same port, viz., the one at the head end of the cylinder, will be opened at the same time and to the same extent. This proves the positions as shown to be correct and illustrates why the eccentric must be moved in the same direction the engine is to run with the *D* valve, and in the opposite direction with the piston valve, in order to secure the same direction of rotation in the engine.

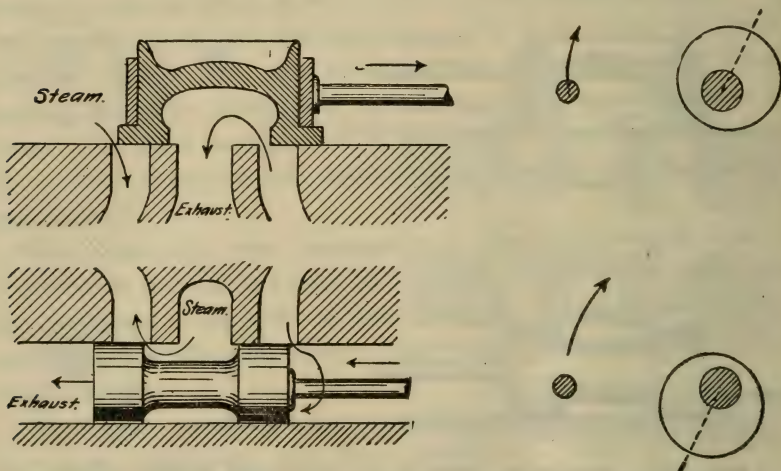


Fig. 161. Valves opening the ports for admission.

When setting valves it is a good plan to obtain as much uniformity of methods as possible, because of the liability to confusion when methods involving different movements of the eccentric are employed. In all the directions that follow it is assumed that the crank is placed on the dead center (see page 195) nearest the cylinder so that when setting the different styles of valves, the same steam port will always be opened first, namely, the one at the head end of the cylinder. The engine, it will be

seen, is thus treated as though it contained but one steam port, which greatly simplifies matters.

In order to show that each particular form of valve of the same type does not require different methods for its proper adjustment, both the simple piston valve and the main valve of the round riding cut-off are illustrated together, the same directions applying to both.

Where marks appear upon the valve stem, or seat, it becomes an easy matter to set a valve quickly and correctly but when these do not appear a different method must be pursued for obtaining them. First remove the chest covers at both ends of the chest

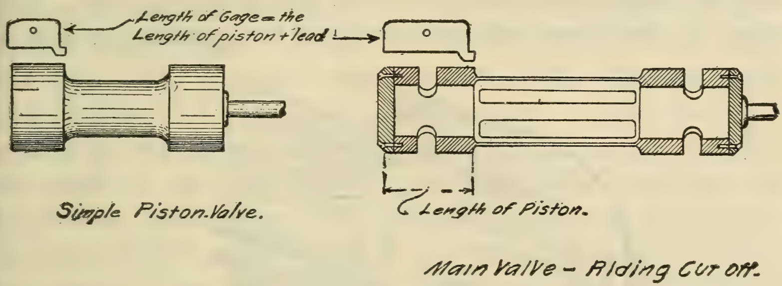


Fig. 162. Templates used in setting piston valves.

and also the valve (both styles) from the chest and lay it upon a clean place on the floor, or bench. Procure a piece of sheet steel about $\frac{1}{16}$ inch thick and file it to the form shown in Fig. 162. Make the length of the gauge thus formed equal to the thickness of the piston on the valve plus the lead, which may be taken as $\frac{1}{32}$ inch. Replace the valve in the chest and connect it to the valve stem. Turn the eccentric from one extreme position to the other and see that the valve opens the ports an equal amount. It is not necessary that the ports be opened exactly wide, the object being to secure exactly the same opening at each end of the valve. If the head end port is opened farther than the other, the eccentric

rod should be lengthened an amount equal to one-half the difference, and should the port at the crank end be opened farthest, the eccentric rod should be shortened a like amount.

Turn the eccentric to the extreme position farthest from the cylinder. Then place the small end of the gauge against the inner edge of the port, and with a scriber make a fine line (a) on the seat as shown in Fig. 163. Remove the gauge, and turn the eccentric in the same direction the engine is to run until the end of the valve reaches the fine line on the seat. Secure the eccentric to the shaft, being careful not to move the eccentric in either direction. Now turn the crank in the direction it is to run until the eccentric reaches the extreme position nearest the cylinder. The gauge is now placed against the edge of the opposite port and a

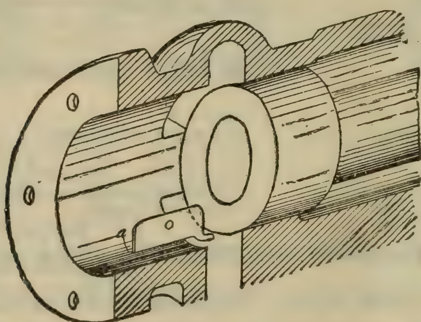


Fig. 163. Showing the use of the template.

fine line drawn on the seat, at the end of the gauge, in the same manner as shown in Fig. 163. Turn the crank to the dead center farthest from the cylinder when the end of the valve should have just reached the line on the seat. If it does not, the crank should be turned sufficiently to enable the distance between the valve and the mark, being measured. The eccentric rod is then to be adjusted so as to move the valve a distance equal to one-half of what the valve lacks of exactly reaching the line on the seat. The valve will then open both ports to the extent of the

lead when the crank occupies the exact dead centers. It is very desirable to have a method of setting the valve without removing the chest covers. By the aid of simple gauges this can be readily accomplished. Take a piece of steel wire and sharpen the ends and bend into the form shown in Fig. 164. With a prick punch make a mark (*b*) on the guide block, place one end of gauge in this mark and make another mark (*c*) where the opposite end of the gauge touches the valve stem. This gauge enables the valve stem being disconnected from the valve stem guide block, and the chest cover put on, and the stem afterward connected up again in exactly the same position (see page 243). Having made this second gauge, place the crank on the exact dead center nearest the cylinder. Then make a prick punch mark (*d*) on the stuffing-box, place one end of the gauge in this mark and then make a second mark (*e*) where the other end of the gauge touches the valve stem. It will readily be seen that when testing the setting of the valve all that is necessary is to place the crank on the dead center nearest the cylinder, then place the gauge in the mark (*d*) on the stuffing-box, and have the eccentric moved until the punch mark (*e*) on the valve stem falls under the point of the gauge. The valve will then have opened the port to the extent of the lead, because it was in this position when the gauge and the marks were first made. If the punch marks are nicely made and not too large the extent of the lead opening may be measured at both ports, by turning the crank to the opposite dead center and making a second punch mark (*f*) on the valve stem by means of the gauge. These two guages should be carefully preserved from injury and from being mislaid so that in case of emergency, such as the slipping of an eccentric, the latter can be returned to its correct position without unnecessary loss of time.

SETTING THE CUT-OFF VALVE.

The following directions are applicable to both the flat slide and the round types of slide valves.

The point of latest cut-off is seldom known exactly by the average engineer because of its unimportance while the engine is in running order, and as this point varies with different engines it is advisable to discard it as an element in valve setting. First place the main valve in its position of mid-travel, that is, place it centrally over the ports. This may be accomplished by finding the center between the punch marks (f) and (e) on the valve stem, bringing the center mark g under the point of the gauge in the manner shown in Fig. 164. The travel of the cut-off valve must first be equalized which is accomplished by turning the cut-off

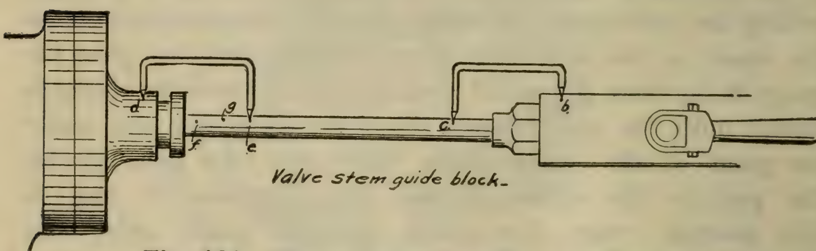


Fig. 164. Trams for setting the cut-off valve.

eccentric to its extreme positions and noting the travel of the cut-off valve over the ports of the main valve. The cut-off eccentric rod should be lengthened or shortened so that the cut-off valve will travel evenly over the ports in the main valve. This, of course, is obtained by measuring the distance from the edge of the ports in the main valve to the ends of the cut-off valve when the latter occupies its extreme positions.

First, assume the engine to have a fixed, or a hand-adjusted cut-off, and that the cut-off valve is to be set to cut off steam at

one-half stroke. Place the crank on the dead center (see page 195) and the full part of the cut-off eccentric the same. Then measure off one-half the length of the stroke from the end of the cross-head as in Fig. 165 and make a light line *f* on the guide. Turn the engine in the direction it is to run until the end of cross-head

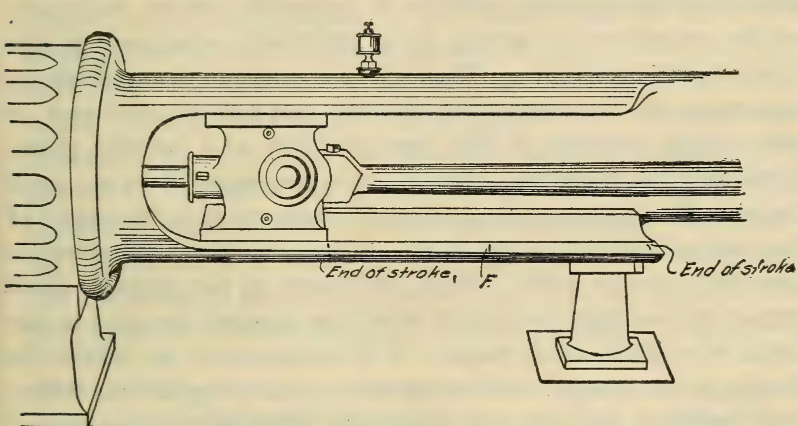


Fig. 165. Method of equalizing the cut-off.

reaches the line *f* on the guide. The piston will now have completed one-half its stroke. Turn the cut-off eccentric in the direction the engine is to run until the cut-off valve opens the port in the main valve wide and just closes the port again in the main valve.

Secure the cut-off eccentric to the shaft at this point. Turn the crank over to the opposite dead center and far enough beyond the center so that the same end of the crosshead will have again reached the line (*f*) on the guide as in Fig. 165. The piston will now have completed one-half of the return stroke and the cut-off valve should have just closed the port in the main valve. If the cut-off valve has moved too far, or not far enough, measure the amount it lacks of just closing the port and then adjust the cut-off eccentric rod an amount equal to one-half the amount of

the discrepancy. The cut-off valve will then close the port in the main valve at exactly the same point in both forward and return strokes.

When an automatic cut-off engine, in which the cut-off eccentric is operated by a shaft governor, first block out the weights to their extreme position or against the stops, the travel of the cut-off valve having been previously equalized in the manner explained above. Then turn the crank to the dead center, preferably the one nearest the cylinder, and turn the full part of the cut-off eccentric to the same position as a starting point. Then turn the eccentric in the direction the engine is to run until the cut-off valve opens the port in the main valve to the extent of the lead or from $\frac{1}{64}$ to $\frac{1}{32}$ inch. Secure the governor wheel to the shaft at this point. Turn the crank to the opposite dead center and see that the cut-off valve has opened the port in the main valve to the same extent. If it has not done so, adjust the length of the eccentric rod an amount equal to one-half the difference between the two lead openings. Take out the blocks and the work will be completed. It will readily be understood that, were the speed of the engine to reach a point, where the governor weights strike the stops, the cut-off valve will admit only steam enough to fill the clearance, which should always be done, because while it does not tend to accelerate the speed it does prevent forming a vacuum in the cylinder, and from drawing in whatever may happen to be in the vicinity of the end of the exhaust pipe. The point of latest cut-off will then take care of itself and will occur at that point for which the valve and gear were designed.

FLAT VALVE RIDING CUT-OFF.

In medium and slow speed engines it is very desirable to have a uniform point of release and constant compression. If the engine is of the automatic cut-off variety the point of cut-off will

necessarily change with each change of load, and if the steam is released, and the point of compression determined by the valve effecting the cut-off, it is plain that as the cut-off varies, the point of exhaust and of compression must also vary proportionately. In order to secure a uniform amount of lead, a constant point of release and of compression, it is necessary that the valve determining these points be given a constant travel. Then in order to produce a variable cut-off a separate cut-off valve must be provided. This is the object of the riding cut-off. The main valve determines the lead, point of release and point of exhaust closure and as the travel of the main valve relative to the crank is unchangeable these functions always remain the same. The duty of the cut-off valve is simply to close the ports in the main valve, and it determines the point of cut-off only. It will be seen, therefore, that with this arrangement of valves, constant lead, exhaust

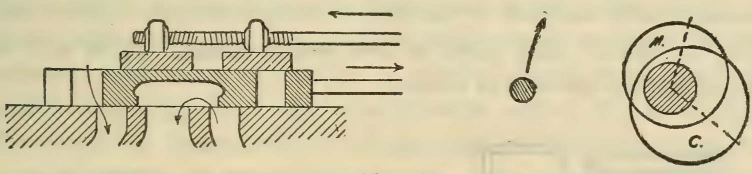


Fig. 166. Riding cut-off — Showing valves and eccentrics.

opening and constant compression are secured while the point of cut-off is constantly changing with the load. Keeping these fundamental facts in mind, it is readily seen that the main valve of the riding cut-off is, in operation, exactly the same as the ordinary *D* slide valve having a fixed travel. In the riding cut-off the travel of the cut-off valve is fixed, so far as length of stroke is concerned, but the times of closing the ports in the main valve are variable and are determined either by hand adjustment or by the governor, depending upon whether the engine is a throttling or an automatic cut-off engine. The points of cut-off are changed by rolling the cut-off eccentric around on the shaft.

The farther the cut-off eccentric is set in advance of the crank the earlier in the stroke will steam be cut off, and, the nearer together the two eccentrics are set, the later will the cut-off occur. The main valve is generally designed to cut off steam at $\frac{2}{3}$ or $\frac{3}{4}$ stroke, so that if the cut-off valve and main valve move together the point of cut-off will be determined by the main valve and will occur at $\frac{2}{3}$ or $\frac{3}{4}$ stroke. Now if the cut-off eccentric (c) be set ahead of the main eccentric (m) as in Fig. 166, it will reach the end of its stroke and start back again before the main eccentric has completed the stroke, thus the cut-off valve moves in one direction and the main valve in the opposite direction and that point in the piston stroke at which the centers of the two valves meet will be the point of cut-off. If the cut-off eccentric be set nearly opposite the main eccentric it is evident that when the main valve reaches one-half of the outward stroke the cut-off valve will have reached nearly one-half of the return stroke and the cut-off will occur at about this point in the piston stroke, which will be approximately one-fourth stroke.

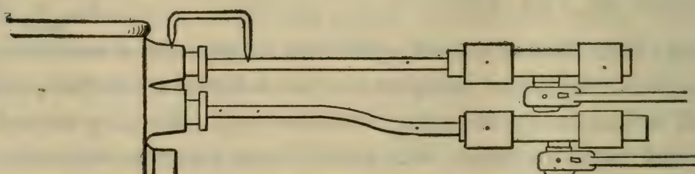


Fig. 167. Setting valves of the riding cut-off.

When setting the valves, first equalize the port opening of the main valve. This is accomplished by turning the main eccentric from one extreme position to the other seeing that both ports in the valve seat leading into the cylinder are opened exactly the same amount. It is not necessary that these ports be opened exactly wide; the object is to see that both ports are opened to the same extent when the eccentric is in its extreme positions.

Having equalized the travel of the main valve place the crank on the dead-center, see page 195, and turn the full side of the main eccentric to a corresponding position. Then turn the eccentric in the direction the engine is to run until the port in the main valve, corresponding to the position of the crank or piston, opens the port leading into the cylinder to the amount of the lead, which may be taken as $\frac{1}{32}$ inch. Now, before moving the engine make a gauge of the form shown in Fig. 167. Put a punch mark on the stuffing-box, and, placing one end of the gauge in this mark, draw a fine line on the valve stem at the opposite point of the gauge. Turn the crank to the opposite dead-center and note the amount of lead opening. If it is not the same as first obtained adjust the eccentric rod to the extent of one-half the difference. Then place the gauge in the punch mark on the stuffing-box and draw a fine line at the opposite point of the gauge.

Turn the crank back again to its first position and note the lead. If it is found to be equal at both ends, apply the gauge again and this time make a light punch mark at the outer point of the gauge. Then put a similar punch mark on the fine line representing the lead at the opposite end of the valve travel. By means of these marks it will be possible to set the main valve correctly without removing the steam chest cover. Now divide the distance between the two punch marks and put a third punch mark at the middle. Turn the crank around in the direction the engine is to run until the middle punch mark falls under the outer point of the gauge. The main valve will now be at the middle of its travel. The travel of the cut-off valve must now be equalized so that the latter valve will travel equal distances beyond the ports in the main valve. This is accomplished in precisely the same manner as with the main valve. Having equalized the travel of the cut-off valve, turn the crank in the direction the engine is to run until the cross-head reaches the point in the stroke at which the cut-off is to occur, which is to be de-

signated by a line drawn on the guide. Now turn the cut-off eccentric in the same direction until it reaches its extreme position. Continue to move the eccentric until the cut-off valve just closes the port in the main valve. Secure the cut-off eccentric to the shaft at this point. Then turn the crank around until the cut-off takes place on the return stroke and see if it corresponds to the point on the previous stroke. If not, adjust the length of the cut-off eccentric rod an amount equal to one-half the difference.

It is important to be able to set the cut-off valve also without taking off the steam chest cover. One punch mark only is required for this. Place the main valve in its position of mid-travel by means of the gauge. Then put a punch mark on the stuffing-box of the cut-off valve stem and, placing the gauge in this mark, put another at the opposite end of the gauge on the cut-off valve stem. See Fig. 167.

This method furnishes a simple and quick means of setting both the main and cut-off valves when an eccentric slips. All that is necessary is to place the crank on the dead-center and bring the proper punch mark under the point of the gauge. Then bring the main valve to its position of mid-travel and with the gauge bring the cut-off valve to its proper position.

The foregoing directions for setting the cut-off eccentric apply to the hand-adjusted gear only. When the cut-off eccentric is operated by the governor, the travel is equalized in precisely the same manner as when hand-adjusted. After equalizing the travel of the cut-off valve, place the crank on the dead-center. The main valve, which is invariably set first, will now open the port, corresponding to the position of the piston to the extent of the lead. Next block out the governor weights against the stops. Turn the full side of the cut-off eccentric to correspond to that of the crank as a starting-point. Then turn the cut-off eccentric (governor wheel), around on the shaft in the direction the engine

is to run until the port in the main valve is opened to the extent of the lead. Secure the governor wheel to the shaft. Turn the crank to the opposite dead-center and see that the cut-off valve opens the port in the main valve to the same extent. If the difference is slight it may be equalized by adjusting the length of the cut-off eccentric rod an amount equal to one-half the difference of the lead openings. Should the difference be great, say, one-half inch, that is, should the cut-off valve lack one-half inch of opening the port in the main valve, it indicates that the cut-off valve is too long, which is apt to be the case where two cut-off valves are employed on the same stem. The valve may be shortened by moving the two parts closer together, moving each part one-fourth of the amount the cut-off valve lacked of opening the port in the main valve. Then begin over again to set the cut-off eccentric and if the adjustments have been carefully made it will open the ports correctly at both ends of the main valve. After fastening the main eccentric and the governor wheel securely to the shaft remove the blocks from the governor weights and the job will be finished.

When the valve gear contains a rocker-shaft of the construction shown on page 322, the eccentric must be turned in the *opposite* direction to that in which the engine is to run, until the main valve opens the port leading into the cylinder, to the extent of the lead.

To Find Average Length of Stroke of Piston.

Revs. per Min.						
25 to 90	Diam. of cyl.	$\times$ 2.5	=	Length of stroke in inches.		
90 " 175	" "	$\times$ 2	=	"	"	"
175 " 300	" "	$\times$ 1.33	=	"	"	"
Single Acting, "	"	$\times$ 1	=	"	"	"

Weight of Engines per Cubic Foot Cyl. Volume.

	Girder Frame.	Box Frame.
Low speed	4800 Lbs.	4400 Lbs.
High speed	5000 "	4800 "
Cross compound non-condensing	8600 "	8200 "
Cross compound condensing	9300 "	8700 "
Tandem compound condensing	8650 "	8204 "
Tandem compound non-condensing	8075 "	7600 "

CHAPTER XII.

THE STEAM ENGINE. — Continued.

Work consists of the sustained exertion of force through space. The unit of work, the foot-pound, is a force of one pound exerted through one foot space. The work done in lifting one pound ten feet, or ten pounds one foot, is ten foot-pounds.

Power is the rate of work, or the number of foot-pounds exerted in a unit of time. The unit of power is the horse-power, and equals 33,000 foot-pounds exerted in a minute, or 550 foot-pounds exerted in a second, or 1,980,000 foot-pounds exerted in an hour. An engine developing fifty horse-power, exerts 27,500 foot-pounds per second, 1,650,000 foot-pounds in a minute. It could raise (friction neglected) 41,250 pounds forty feet in one minute.

A belt running over a pulley at 4,000 feet per minute, pulling with a force of 240 pounds (fair load for a 4-inch belt) will transmit

$$\frac{240 \times 4,000}{33,000} \text{ equal thirty horse-power (nearly).}$$

If moving at 1,100 feet per minute, the result would be

$$\frac{240 \times 1,100}{33,000} \text{ equal eight horse-power.}$$

A gear-wheel, the cogs of which transmit a pressure of 1,800 pounds (fair load for $1\frac{1}{2}''$ pitch $6''$ face) to the cogs of its mate, the periphery velocity of the wheels being ten feet per second, transmits

$$\frac{1,800 \times 10}{550} \text{ equal thirty-three horse-power nearly.}$$

If speed was 360 feet per minute, it would transmit

$$\frac{1{,}800 \times 360}{33{,}000} \text{ equal twenty horse-power nearly.}$$

The horse-power developed by a steam engine consists of two primary factors, *Piston Speed and Total Average Pressure* of steam upon the piston.

Piston speed depends upon the stroke of engine and the number of revolutions per minute. An engine with stroke of twelve inches, making 300 revolutions per minute, has a piston speed of

$$\frac{2 \times 12 \times 300}{12} \text{ equal 600 feet per minute.}$$

Piston speed of an engine with 24″ stroke at 150 revolutions per minute:

$$\frac{2 \times 24 \times 150}{12} \text{ equal 600 feet per minute.}$$

Total average pressure depends *on area of piston and mean effective pressure* per square inch exerted on piston throughout stroke. The mean effective pressure (M. E. P.) in any case can only be accurately obtained by means of the steam engine indicator, and depends upon the load engine is carrying.

GENERAL PROPORTIONS.

Diameter of steam pipes:

Slide-valve engine, $\frac{1}{4}$ diameter of piston.

Automatic high-speed engines, $\frac{1}{3}$ diameter of piston.

Corliss engine, $\frac{3}{10}$ diameter of piston.

Diameter of exhaust pipes:

Slide-valve engine, $\frac{1}{3}$ diameter of piston.

Automatic high-speed engine, $\frac{3}{8}$ diameter of piston.

Corliss engine, $\frac{1}{3}$ to $\frac{3}{8}$ diameter of piston.

	Displacement of piston in one stroke.
Clearance spaces:	
Slide-valve engine	0.06 to 0.08
Automatic high-speed engine, single valve	0.08 to 0.15
Automatic high-speed engine, double valve	0.03 to 0.05
Automatic cut-off engine, Corliss type, long stroke	0.02 to 0.04

Weights of engines per rated horse-power:

Slide-valve engine	125 to 135 lbs.
Automatic high-speed engine	90 to 120 lbs.
Corliss engine	220 to 250 lbs.

Fly-wheels, weight per rated horse-power:

Slide-valve engine	33 lbs.
Automatic high-speed engine (According to size and speed.)	25 to 33 lbs.
Corliss engine (According to size and speed.)	80 to 120 lbs.

RULES FOR FLY-WHEEL WEIGHTS, SINGLE CYLINDER ENGINES.

Let d = diameter of cylinder in inches.
S = stroke of piston.
D = diameter of fly-wheel in feet.
R = revolutions per minute.
W = weight of fly-wheel in pounds.

For slide-valve engines, ordinary duty . $W = 350{,}000 \frac{d^2\ S}{D^2\ R^2}$

For slide-valve engines, electric lighting. $W = 700{,}000 \frac{d^2\ S}{D^2\ R^2}$

For automatic high-speed engines . . $W = 1{,}000{,}000 \frac{d^2\ S}{D^2\ R^2}$

For Corliss engines, ordinary duty . . $W = 700{,}000 \frac{d^2 S}{D^2 R^2}$

For Corliss engines, electric lighting . $W = 1{,}000{,}000 \frac{d^2 S}{D^2 R^2}$

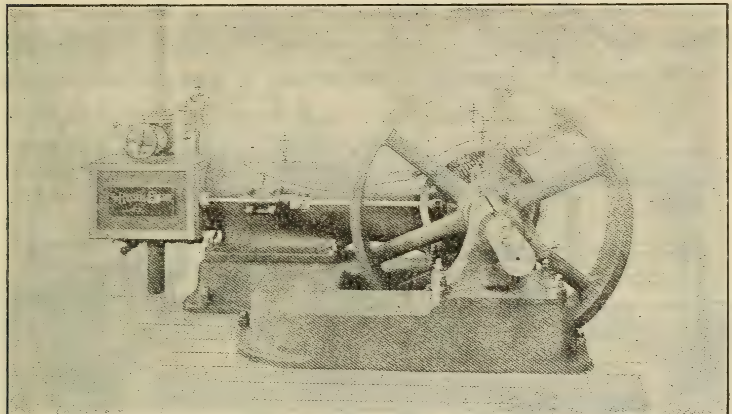


Fig. 168. The Russell engine.

SETTING THE VALVE ON A RUSSELL ENGINE, SINGLE VALVE TYPE. THE SAME PRINCIPLE LAID DOWN IN THE SETTING OF THE COMMON SLIDE VALVE MUST BE ADHERED TO.

The style of valve is shown in cut, Fig. 169. It is, to some extent, a moving steam chest with the steam all within itself, admitting only enough steam into the chest to keep the valve to its seat, against the maximum tendency to leave it. This pressure in the chest is found with the valve as at present proportioned, to be about 45 per cent of that contained within the valve. The cut shows the valve and section of cylinder so plainly as to render any detailed explanation of same almost unnecessary.

The eccentric operating the valve is under control of the governor, as shown in cut Fig. 170, which regulates the speed of the engine by sliding the eccentric across the shaft, either forward or backward, as the weights change their position, thereby cutting the steam off earlier or later in the stroke, as the governor, or more properly, the weights adjust themselves to the load.

When the eccentric is moved across the shaft in a direction that reduces its eccentricity, the steam is cut off earlier in the

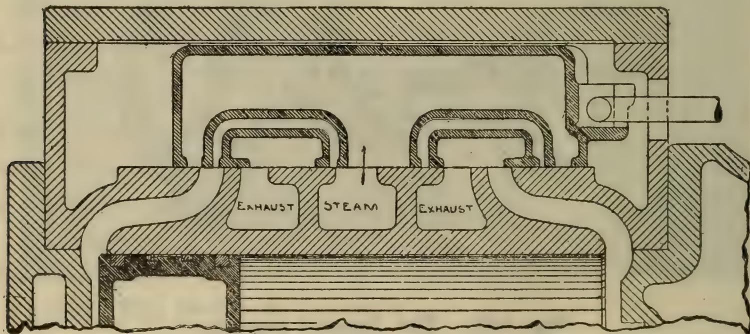


Fig. 169. Sectional view of Russell steam valve.

stroke; when the eccentric is moved in the opposite direction, the steam is cut off later in the stroke. The extreme range of this cut-off is from 0 to $\frac{3}{4}$ of the engine's stroke, and this whole range of adjustment is under complete control of the governor.

To preserve a certain determined speed with the smallest possible variation, as changes occur in the load or pressure, is the function of the governor. The cut-off must always be proportioned to the load. When the engine is running empty, the steam is cut off at the beginning of the stroke and the governor weights are at their extreme outer position. With a heavy load, steam follows farther and the weights are nearer their inner position. Be-

tween these two limits, any number of positions of the weights, and corresponding angular positions of the eccentric, may be had ; and

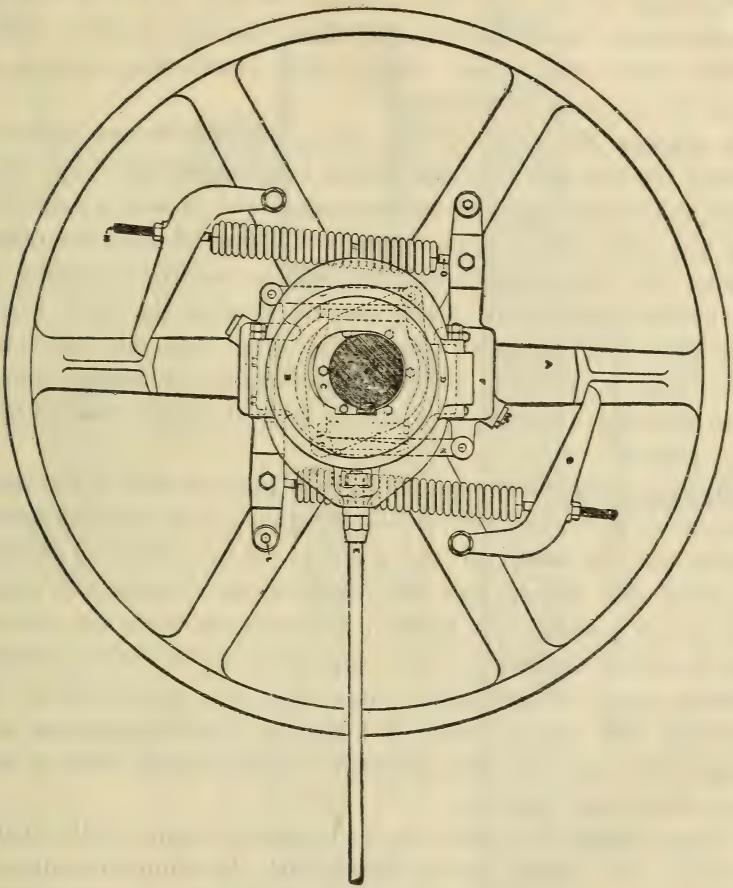


Fig. 170. The Russell shaft governor.

as the steam is thus adapted to the load in each position, it follows that a slight increase or decrease in speed must make a change in the cut-off and bring the engine again to standard speed.

In setting the valves it is necessary to mark the ports in the valve face at the outer edge of the steam chest, and also to mark on the back of the valve the ports in its face, so that it may be adjusted after being placed in the chest, in which position it presents a blank surface that, without these marks, would afford no means for knowing its position.

In placing the valve in the chest, see that it fits perfectly against the seat and that the bottom bearing, on which the valve rides, is at right angles to the valve seat, and in such a condition that the valve will not be tipped away from its seat, but rather against it. This latter condition will be insured by easing off the bottom strip at the inner corner, so that the valve would bear hardest at the outer edge. The hinge nut, into which the valve stem is screwed, as well as its trunnion bearings, should fit so that the valve lays closely to its seat, rather than be held away from it.

Having extended the marks of the ports as well in the valve seats as in the valve itself, to the outside, it now becomes necessary to get the center of the travel of the eccentric and connect the valve and rod, so that the valve will travel equally on either side of this center. The throw of the eccentric leads the crank in the direction the engine runs, and with the eccentric properly located, as it cannot help being, because it is attached to the governor and the governor is keyed to the shaft, the lead will remain the same with the governor weights in their outer as well as in their inner positions.

These valves are usually marked with the engine on the center at either end, marks corresponding with the admission edges of valve and seat. The hinge nut connection makes it convenient to examine these valves without disconnecting or disturbing any adjustments made. The valve rod has right and left-hand threads for adjustment, and final adjustment can be made without taking off the steam chest cover.

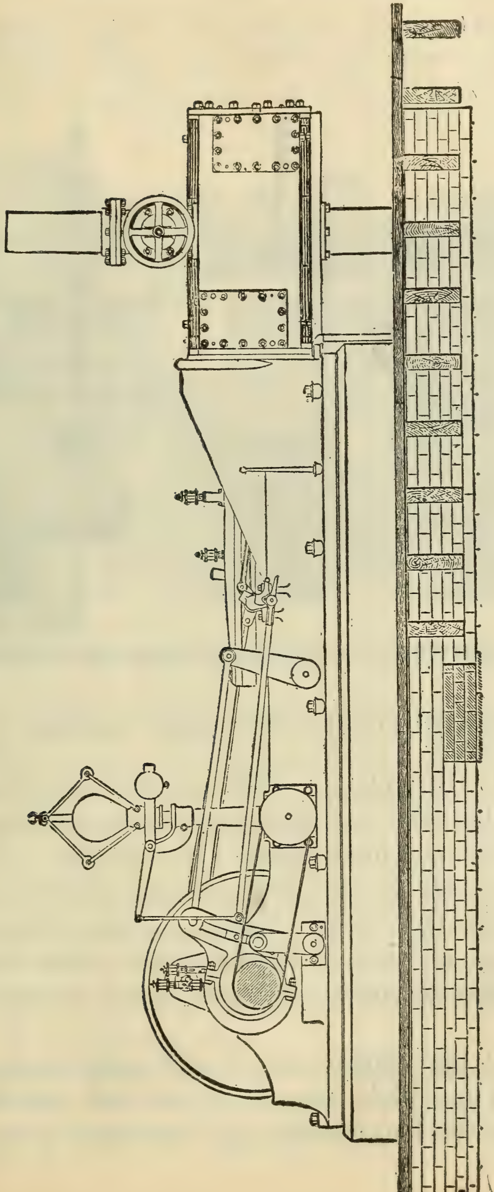


Fig. 171. The Porter-Allen engine.

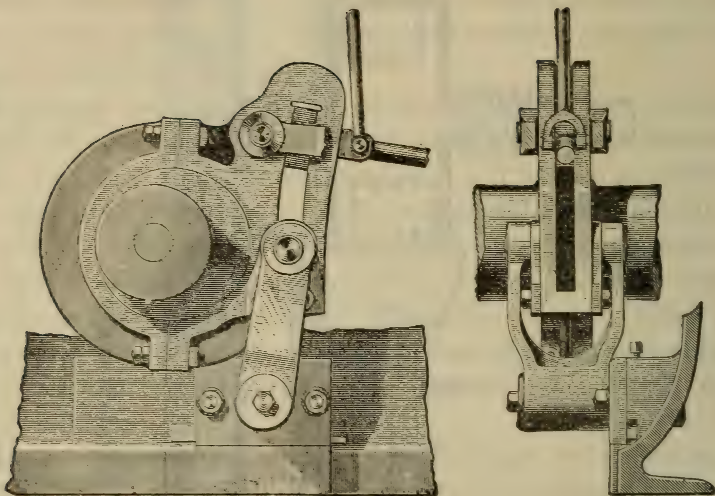


Fig. 172. Governor connections — Porter-Allen engine.

THE ADJUSTABLE PRESSURE PLATES.

Description of these plates. — The construction of these pressure plates and the method of adjusting them are fully represented in the sections of the cylinder, Figs. 173 and 174.

On the lower side of the horizontal section Fig. 173, both admission valves are shown, working between their opposite parallel seats, one of which is formed on the cylinder, and the other on the pressure plates, the latter having cavities opposite the ports.

The valve at the farther end of the cylinder is at the extremity of its lap, while the one at the crank end has commenced to open the four passages for admission of the steam.

The vertical cross-section, Fig. 174, passes through the middle of one pressure plate and shows its form and the means employed for its adjustment. It is made hollow and most of

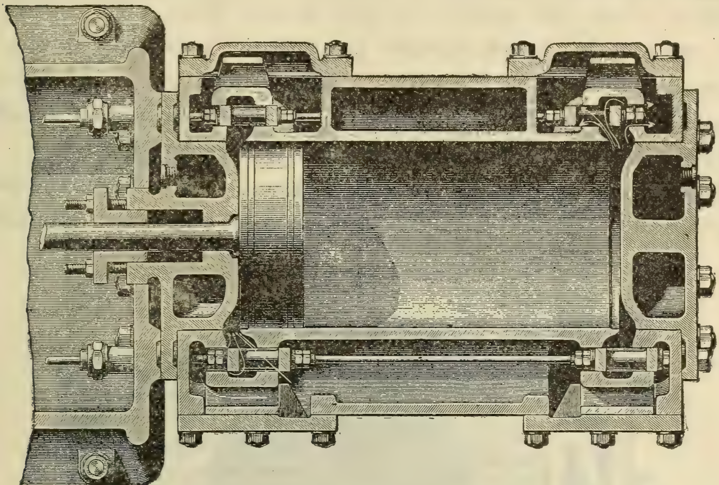


Fig. 173. Cylinder and valves — Porter-Allen engine.

the steam supplied to two of the openings passes through it. It is arched to resist the pressure of the steam without deflection. It rests on two inclined supports, one above and the other below the valve. These inclines are steep, so that the plate will be sure to move freely down them under the steam pressure, and also that it may be closed up to the valve with only a small vertical movement. It is prevented from moving down these inclines by a screw, passing through the bottom of the chest, the point of which, as also the plug against which it bears, is of hardened steel.

The pressure plate is held in its correct position by projections in the chest, on one side, and tongues projecting from the cover on the other, which bear against it near each end, as shown.

Between these guides, it is capable of motion up and down its inclined supports, and also directly back and forth between the valve and the cover.

The pressure of steam is always on this plate, and tends to force it down the incline to rest on the valve. By means of the screw it is forced against the steam pressure, up the inclines and away from the valve. This adjustment is capable of great precision, so that the valve works with entire freedom between its opposite seats, and still is steam-tight.

How these plates act as relief valves. — Whenever the pressure in the cylinder exceeds that in the chest, the admission

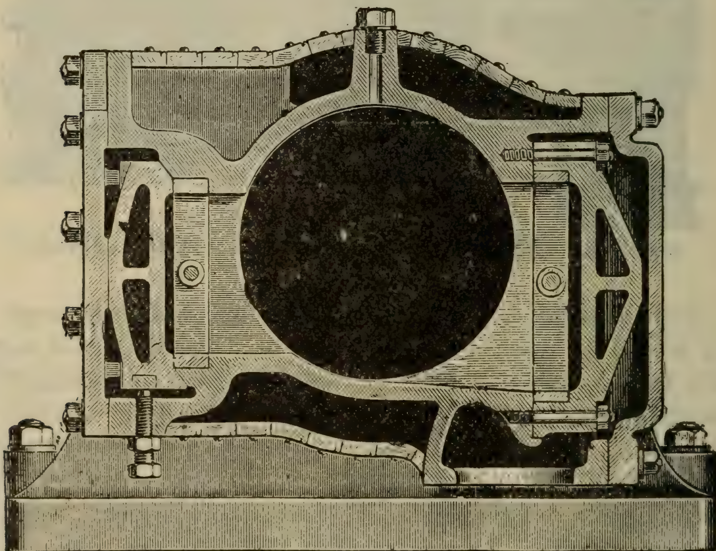


Fig. 174. Cross-section of cylinder and valves.

pressure plate is instantly moved back to contact with the cover, thus affording an ample passage for the discharge of water before it can exert a dangerous strain. This plate is superior

in this action to any of the ordinary forms of relief valve, both in the area opened, and also in being self-adjusted to the pressure, and opening fully the instant that is exceeded.

How to keep the admission valves tight.—These valves, though moving in complete equilibrium, are liable to slight wear.

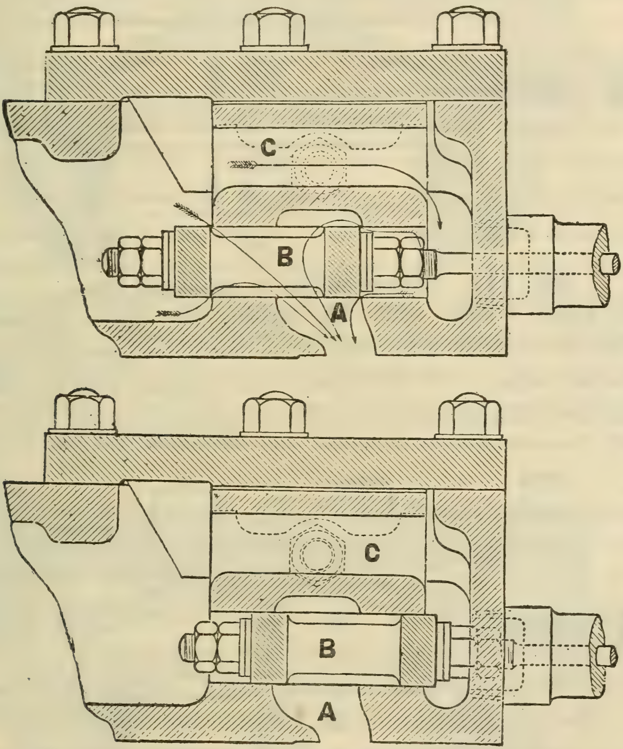


Fig. 175. Section of Porter-Allen steam valve.

This should be taken up as it appears, by letting down the pressure plates. The construction of these plates and the method of adjusting them, are shown in the accompanying sections, made through the steam chest at one end of the cylinder. Of these, the drawings are horizontal sections, showing the four-openings of

the valve — first, when commencing to open, with arrows indicating the course of the steam; and, second, at the extreme point of its lap; while Figs. 176 and 177 are vertical sections, showing the

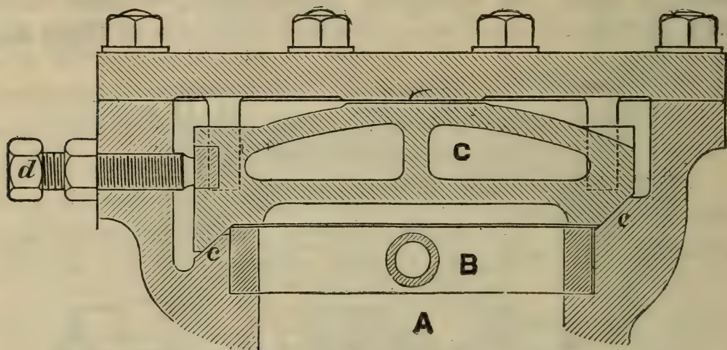


Fig. 176. Cross-section of pressure plate.

pressure plate — first, when by turning the bolt *d* forward it is forced up the inclines and away from the valve, producing a leak; and second, when it is let down to its proper working position. *A* is the port, *B* the valve, and *C* the pressure plate. The latter

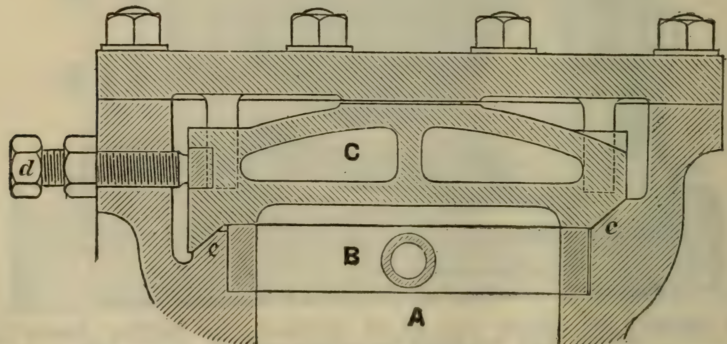


Fig. 177. Porter-Allen valve and pressure plate.

is made with a trussed-back and so cannot be deflected by the steam pressure. Through the passage thus formed, the steam reaches two of the openings.

The pressure plate rests on two inclined supports, *c*, *c*, and the pressure of the steam forces it down these inclines as far as the bolt *d* underneath will allow. This bolt holds the plate just off from the valve, so that the latter moves freely, and is still steam tight. Whenever leakage appears, a minute turning of this bolt backwards lets the pressure plate down and closes it.

Provision is made for readily detecting the least leakage, as follows: When the engine is warmed up in its normal working condition, open the indicator cocks, or in the absence of these, remove the plugs from the top of the cylinder, unhook the link rod, and set the valves by the starting bar so that both ports are covered, and turn on the steam. If the valve leaks at the end of the cylinder, which is not then open to the atmosphere or the condenser, the steam will blow out at the opening provided, having no other outlet. Then let down its pressure plate by backing the bolt very carefully till the leak disappears. The valve should still move freely when the leak has disappeared, and the pressure plate must not be let down any closer than is necessary for this purpose.

Leakage at the opposite end of the cylinder will not generally be seen, the steam escaping freely by the open exhaust. To test its valve in the same manner, the engine must be turned on to the opposite stroke. These examinations should be made from time to time.

In the small engines, which have no starting bar, the valve rod can be disconnected and moved by hand to test this point.

An engine should never be started till it is warmed up. The valves warm quicker than the supports on which the pressure plates rest, and are tight between their seats by expansion, until the temperatures have become nearly equalized. Provision for detecting and stopping any leak of steam is the crowning excellence of this valve.

DIRECTIONS FOR SETTING THE VALVES OF THE PORTER-ALLEN ENGINE.

To set the admission valves. — Place the engine on one of its dead centers as explained on page 195. Then raise the governor, bringing the center of the block between the centers of the trunnions of the link.

With the governor remaining up, set the valve that is about to open, giving to it a lead of from $\frac{1}{16}''$ to $\frac{3}{16}''$, according to the size of the engine. High speed requires considerable lead. Repeat this for the other valve on the opposite center.

On letting the governor down, the crank remaining on the dead center, it will be seen that the valve is moved a short distance. This motion of the valve, produced by moving the block from the trunnions to the extremity of the link while the crank stands on the center, is the same in amount on either center and takes place in the same direction; namely, towards the crank. Its effect is, therefore, to cover the port nearest the crank and to enlarge the opening of the port farthest from it; so that the lead, which is equal at the earliest point of cut off, is at the crank end of the cylinder gradually diminished, and at the back end increased in the same degree as the steam follows farther.

The effect of this is to equalize the opening and cut-off movements, so that, on setting the governor at any elevation whatever and turning the engine over, the openings made and the points of cut-off will be found to be identical on the opposite strokes, from the commencement up to the maximum admission. This difference in the lead is also singularly adapted to the difference in the piston velocity at the two ends of the cylinder.

In case the indicator shows that the lead of either admission valve requires to be changed, this is done without opening the chest, by lengthening or shortening the stem at the socket of

its guide, bearing in mind that each valve moves towards the middle of the cylinder to open its port.

To set the exhaust valves. — These have an invariable motion, and are admirably adapted to their purpose. They are set so as to open before the end of the stroke enough to give ample lead, and close again when the piston is on the return stroke, early enough to effect the required compression.

All the valves are held between pairs of brass nuts, of which the inner one is flanged. These nuts must be securely locked, and should be so set upon the valve that it is free to adjust itself between the nuts while yet sufficiently tight that no "lost motion" exists. To avoid the consequences of a mistake, care should be taken, before closing the valve chests, to turn the engine slowly through an entire revolution, while the movements of the valves are carefully watched, so as to insure that they have not been so set as to bring the valves or their nuts into contact with the ends of the chest at the extremes of their movements.

The governor. — The Porter governor, original in its type, stands unexcelled as adapted to stationary engines, requiring close regulation. The active parts are very light, the power being derived from a high rotative speed, causing a sensitiveness in its movements that will arrest fluctuations and produce uniformity in the running of the engine. It has been so perfected that at the present day it is easily adapted to the requirements of any class of work necessitating a governor, and is especially desirable for an engine where a steady speed is necessary.

The speed of this governor being constant, makes it equally efficient upon an engine running either at a high or low number of revolutions. That is to say, the speed of engine can be altered from time to time by changing the governor pulley, the governor itself continuing to run at the same speed and under the same strains, and being stationary, it is always open to observation.

The Armington and Sims engine, as is well known, is of the high speed type, and in its earlier form was designed with double eccentrics, one inside of the other. These eccentrics are operated by the shaft governor, and the compound motion produced by the movements of the two eccentrics is such that the valve has equal lead for all points of cut-off.

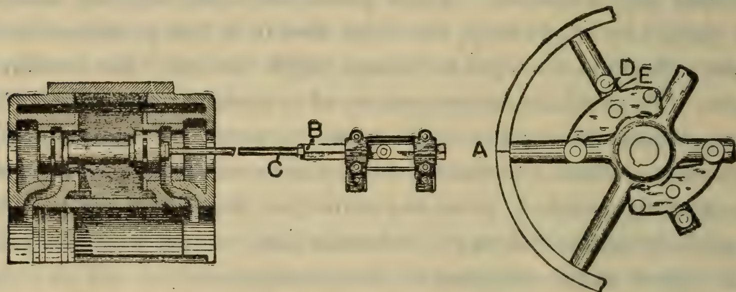


Fig. 178. Valve gear of the Armington & Sims automatic engine.

The method of setting the valve is very simple, for all engines of this make are sent out with the valve stem and slide marked at points *C* and *B* in the sketch, and these points should be set just three inches apart. The following are the directions which the builders supply: —

"If the distance between *B* and *C* is just three inches you will know that the valve is all right. If, however, you wish to put in a new valve and adjust, then remove the steam-chest cover and place the engine on the center as follows: Place line marked *A*, which is on the crank pin side, with line on opposite side of rim marked *F* (not shown in drawing), level with engine; now take out, or loosen up the springs and block the weights out so that the distance between weights and pin at *D E* will be $\frac{3}{4}$ of an inch; adjust the valve-stem at the guide so that by turning the engine over

from one center to the other the lead will be the same at both ports; then make a new mark distinctly on the valve-rod, so that the distance B C will be the standard three inches. See Fig. 178.

"It is not possible to reverse the direction of running without sending to the factory for new parts. The governor is not constructed so that one set of parts can be used for running both ways."

THE CARE AND MANAGEMENT OF HARRISBURG ENGINES.

It is essential to the successful operation of any high-class and expensive machinery, that the person in charge be gifted with a fair degree of intelligence and alertness, and while it is not attempted to formulate new rules as a guide to the person in

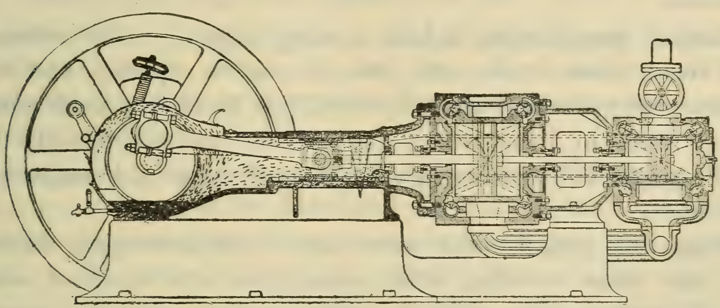


Fig. 179. Sectional elevation of Harrisburg standard four-valve tandem compound engine.

charge of an engine, the fact must not be overlooked that a great deal depends upon the skill and judgment of the operator himself, and that it is manifestly impossible to give rules other than of a general character and which may frequently have to be modified to suit the different conditions that may arise. However, the

following are some suggestions for the convenience of operating engineers: —

When engines of these styles have been properly erected, the steam, exhaust and drain connections completed, and the piston and valve rods packed, the operator should be careful to see that all parts are in proper position and firmly secured.

The bed should be thoroughly cleansed inside and a good quality of machine oil poured into the reservoir beneath the crank, until it is just in contact with the crank disc.

A mineral oil only should be used, and of medium viscosity. Fill the eccentric lubricating cup and flush the main bearings with the oil.

The cylinder lubricator should be filled with a first-class quality of cylinder oil, of heavy body.

The best oils obtainable are the most economical, without question.

Careful preparations before starting engine. — The cylinder and steam chest drain valve should now be opened, and the throttle valve carefully started just enough to allow a small quantity of steam to flow through the cylinder and out through the drain pipes, but not enough to actually start the engine in motion.

After the cylinder and valves have been thoroughly heated and any water standing in the steam pipes thus blown off, start the oil flowing in the cylinder lubricator cup. A general survey of the engine should now be taken and if everything is found to be in proper condition, carefully open the throttle valve and bring the engine gradually up to speed, when it should be noted that the governor is controlling the machine. Examine the bearings and eccentric to see if the oil is flowing properly, and make sure that every part is operating smoothly, after which the drain valves may be closed.

Adjustments for wear. — When the engine has been in opera-

tion long enough to necessitate the adjustment of the working parts, care should be used to avoid adjusting them so close as to cause heating, and the following general rules should be observed: —

The caps on the main bearings should always have sufficient liners underneath to enable the nuts on the bearing studs to draw the cap down solidly upon them and not pinch the shaft, which should be free to revolve in its bearings without unnecessary play.

Adjustment of crank-end connecting rod. — In adjusting the connecting rod box at the crank pin end the same general rules should be observed regarding the liners under the cap, the large nuts drawn solidly upon it, the small nuts firmly jammed, and the cotter pins placed in position.

The adjustment of the box should then be tested with a lever about 12 inches in length, the adjustment being so made that with a lever of this length the operator can easily move the end of the connecting rod sufficiently to take up the side play between the flanges on the crank pin and the ends of the box. The adjustment should never be made so close that this side movement cannot be observed.

Adjustment of cross-head pin box. — The adjustment of the connecting rod box at the cross-head pin end should be made by removing the name plate from the engine frame and placing the crank on the center nearest the cylinder, then with the wrench provided for that purpose, slack off both wedge screws at the upper and lower sides of the connecting rod, and draw the wedge up until it is solid against the box, then slack off that screw about a sixth of a turn and draw up the other so as to firmly lock the wedge; this method prevents the box from pinching the cross-head pin.

The "flats" on the cross-head pin should always be at the top and bottom to avoid wearing a shoulder, and the nut on the end should be drawn up firmly, but not so much as to spring the

bosses of the cross-head together, nor yet enough to make the box tight on the ends.

Proper adjustment of cross-head in the guides is made by liners of paper or tin, placed between the bronze shoes and the body of the cross-head.

Adjustment of cross-head shoes. — In order to do this it is necessary to remove the pin and the end of the connecting rod from the cross-head, and with a wooden lever placed in the pin hole turn the cross-head until the shoes are out of the guides, then remove the shoes and place the liners beneath them. Care should be used that the cross-head does not fit the guides too closely, and that it can be moved freely with a short lever from one end of the guides to the other, while disconnected from the connecting rod.

The cross-head should never be run very close and should always be free enough to allow long and continuous runs without causing the top of the bed over the guides to feel uncomfortably warm to the touch.

Attachment of cross-head to piston rod. — When making any adjustments of the cross-head, it is well for the operator to assure himself that the lock nut, which prevents the piston rod from turning in the boss at the end of the cross-head, is securely in place. All but the largest Harrisburg engines are tested under steam before leaving the works, and the valves set with the indicator.

The distance from the cylinder head end of the valve, when the crank is on the center nearest the cylinder, is marked on the end of the cylinder directly underneath the steam chest cover. If from any cause the valve should become deranged, place the crank on the center described and with a scale or rule, see that the valve position corresponds to the dimension marked on the end of the cylinder; and if out of position, it can easily be re-adjusted by means of the device provided for that purpose, at the outer end of the valve stem.

On the Harrisburg Ideal engines, where the ball joint connection is used between the valve stem and the eccentric rod, the wear is followed up by filing the end of the bronze connection that the cap is screwed against, which holds the ball in place. And on the Harrisburg Standard engines, where the ram box connection is used, the adjustment is made by filing the half of the bronze box, which is attached to the end of the eccentric rod that connects with the ram.

Adjustment of eccentric strap. — The eccentric strap adjust ment is made by liners placed between the halves of the strap and double nutted bolts. When adjustment is necessary, the other end of the eccentric rod should be disconnected and after drawing up the strap bolts it should be tested by giving the strap a half revolution about the eccentric. If it is found that the friction between the strap and eccentric is sufficient to support the weight of the rod, the bolts should be loosened until the strap moves freely without lost motion. The double nuts should then be locked and the cotter-pins replaced in the ends of the bolts.

How to alter engine speed. — The governor used on all Harrisburg engines is the centrally balanced centrifugal inertia type. A few words of explanation may be of service to operating engineers.

The weight arms are constructed with differential weight pockets, to allow of a considerable range of speed adjustment without altering the tension of the springs. If an increase in speed is desired, remove weights of an equal thickness from the weight pockets of the levers, and add weights of an equal thickness to obtain a decrease in speed. If an increased speed causes the governor to "race" or "weave," move the clamp in the slot, to which the outer end of the spring is attached, farther from the small end of the weight lever. If this does not entirely correct this sensitive condition, screw the plug into the spring until the racing ceases. If the decrease of speed so obtained renders the governor too sluggish in action, move the clamp in the

slot in the opposite direction. If this does not improve the regulation, and the speed is lower than desired, add weights of an even thickness, increasing the spring tension until the proper speed is obtained. The main lever bearings, which are equipped with anti-friction steel rollers, should be oiled about once a week, and taken out and cleaned about once a month; the other joints fitted with compression grease cups, should be treated in the same manner. About once a month, also, the springs should be disconnected and the governor and valve gear tested by hand, to make sure all joints are working freely.

The foregoing will apply also to the Harrisburg Standard and Ideal compound engines, and, in general, to the Harrisburg self-oiling four-valve engines. Adjustment for wear in the valve gear connection of the latter type of engines is obtained by filing the halves of the bronze boxes on the ends of the rods connecting the valves with the wrist-plates and rocker arms, and on the wrist-plate and rocker arm pins, by means of bronze shoes let into the sides of the bearings, the wear being followed up by the screws provided with lock-nuts, and all bearings lubricated by means of compression grease cups. The Harrisburg Corliss engines,of the larger sizes, are provided with quarter-boxes in the main bearings with wedge and screw adjustment, and are built self-oiling or otherwise, according to size. The lubrication of the principal bearings is accomplished by means of oil cups, and the valve-gear connections by means of conveniently arranged grease cups.

McINTOSH AND SEYMOUR HIGH SPEED ENGINE.

How to set the valve. — When the engine is sent out from the shop, the valves are set and trammed with three-inch tram from the valve rod to the valve rod slide at *C D*, and from the eccentric rod to the eccentric rod head at *E F*, on the valve slide end, and a tram is furnished with the engine, or a new tram can be made with exactly three inches distance between the points, which will suffice.

In case the tram marks become lost, or, owing to wear of the valve gear, the length of connection is altered, the proper procedure is to put the engine on one center, and then on the

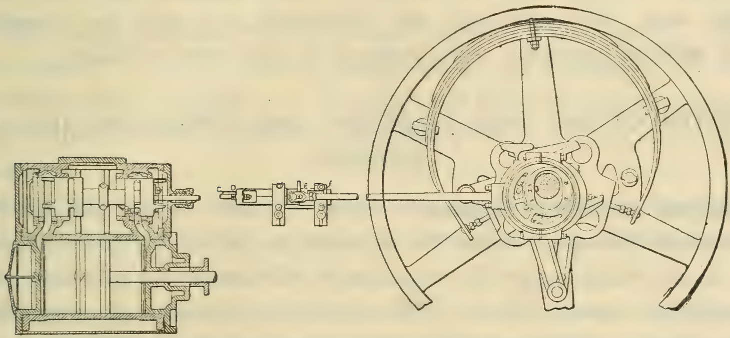


Fig. 180. A sectional cut of McIntosh and Seymour high-speed engine, showing valve and governor.

other, and observe the leads which occur when the governor is in the normal position of rest. See Fig. 180. The lead on the crank end should be three times as much as the lead on the head end, if the connection between the valve and eccentric is of proper length.

When the valve is set this way, the cut-off on the two ends of the cylinder will be approximately equal at one-quarter cut-off on the smaller size engines having inside governors.

Preliminary to adjusting connections between the valve and eccentric, care should be taken that the mark on eccentric *G H*, corresponds to the mark on the pendulum.

In examining the steam leads, as described above, it should be noted that the surface *B* on the valve has nothing to do with the steam distribution, but it is merely to give ample wearing surface, and that the steam is admitted to the cylinder through the port which is between *B* and the steam edge, which is at *A*, and the lead should be measured between this steam edge and the

edge of the port leading to the cylinder. On engines of larger size having outside governors, a similar method should be employed in setting the valves, except that the trams are four inches from point to point, and should be used between the valve rod slide and valve rod, and the eccentric rod and the eccentric rod head at governor end, instead of slide end, as above.

INSTRUCTIONS FOR STARTING AND OPERATING IDEAL ENGINES.

Before starting engine. — Open cylinder cocks and throttle valves sufficiently to warm the cylinder and valve. Place sufficient oil in the basin under the crank so it will stand one inch above the bottom of crank discs. When receiving a new engine from the shops with visible stuffing-box and water drain, before filling the crank case with oil, previous to starting, pour water in opening

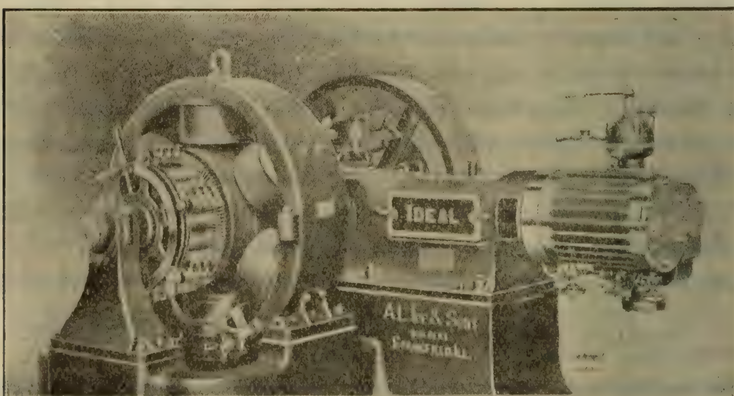


Fig. 181. The Ideal high speed engine.

in frame into pocket under piston rod stuffing-box, until water overflows through trap connected therewith attached to outside of frame. Fill cylinder lubricator and start it to feeding. Fill oil

pump, and pour engine oil into pocket on main bearings. Fill eccentric oiler and start it feeding. After the steam chest and cylinder are warm, turn the engine over by hand to see that all is free and right to start.

Open the throttle valve gradually, *start engine slowly.* After the engine is up to speed, pump five or six strokes of oil into cylinder with oil pump. The oil should flow in streams through both pipes on the crank cover into the pockets of the main shaft bearings.

This oil passes from the main bearings through the crank pin and is distributed over cross-head pin and slides. Occasionally clean out the oil passages in crank pin.

Supply, as needed, a little fresh oil to the basin, and if the oil in the engine bed becomes thick, gritty or dirty, so as not to flow freely through oil passages, draw it off and replace with fresh oil. Filter the old oil and use it over continuously. Use a pure mineral oil that will not thicken by the churning it receives.

Serious damage and cutting of the cylinder and valve will result from allowing the lubricator to cease feeding, even for a few minutes. If the engine is a new one from the shops, feed plenty of oil through the lubricator and oil pump for the first few weeks after starting. Use one drop of oil per minute for each ten horse-power, or ten drops per minute for 100 horse-power engine, for the first thirty days; after which, one-half this amount will be sufficient, if the oil is of good quality. If the boiler is priming or foaming, use double the quantity of oil to protect the cylinder and piston from cutting. A little graphite fed into cylinder is very beneficial.

The governor. — Fill the cups on governor bearing with grease and give the cap $\frac{1}{4}$ turn every day. Screw the cap to the stuffing-box on dash-pot loosely, only using the hand to turn the cap. The governor should be taken apart every two or three months and bearings cleaned with coal oil to remove gum. If governor

has a dash-pot, it should be refilled with glycerine once or twice a year. Oil may be used in the dash-pot in place of glycerine, unless the engine is in a cold room where the oil is liable to congeal. To refill dash-pot, unscrew cover on end.

In taking the governor apart, allow the sliding block, which holds the end of the governor spring, to remain with its outer edge on a line with a mark across the face of the slide, and in re-adjusting the spring, place the same tension on it as before, which can be ascertained by measuring the length of the thread through the nuts before slacking up the spring. If trouble is had with springs breaking it is because of their being worked under too much tension. The speed of the governor is changed by moving the weight on the lever.

To increase the speed of the engine, move the weight on the governor lever near to the fulcrum pin. To reduce the speed, move the weight out toward the end of the lever. Tightening the spring will also increase the speed, but will cause the engine to "race," unless at the same time the block, which holds the end of the spring, is moved toward the center of the wheel. The proper way to change the speed is by moving the weight, allowing the spring to remain in its marked position.

Moving the block, which holds the spring, towards the rim of the wheel, will make the governor more sensitive and regulate more closely; but if moved too far, this will cause the governor to "race." Moving the block towards the hub of the wheel has a tendency to stop the "racing," but if moved too far the speed of the engine will be reduced with the increased load. If any of the bearings of the governor bind, or require oiling or cleaning, the governor will "race." These bearings should be kept clean and in good condition and the stuffing-box to the dash pot must not be screwed up tight, as that will cause the governor to "race" when set for close regulation.

The face of the slide is marked with a line where the outer

edge of block which holds the spring should be. Figures stamped on the face of the slide, give length of end of eye-bolt extending through nuts. This gives the right tension to the spring. Tightening the spring will give closer regulation, but will cause the governor to " race " if the spring is too tight. " Racing " caused by over-tension of spring, can be stopped by moving block nearer to center of wheel.

To set valve. — When necessary to ascertain if the steam valve is properly set, proceed as follows: Take off the cover or elbow on outer end of steam chest, so access can be had to end of valve. Turn the engine over until the valve has traveled as far as it will go towards end of steam chest. Then measure from the end of steam chest to the end of the valve, and this distance should be represented by the figures in inches and fractions on end of steam chest. If measurements do not agree, set valve by screwing the valve stem at the ball joint.

Square, braided flax packing is the best kind for piston rod and valve stem. Don't screw the glands up tight; allow them to leak a little. The valve stem has only exhaust steam — don't pack it tight. Screw it up by hand only. Screwing the piston rod gland up tight may cause the piston to thump or pound the cylinder, and heat and cut the piston rod.

Safety caps. — The safety caps attached to drip valve under the cylinder are intended to break, in order to save damage to the engine if water enters cylinder. They will protect the engine from breaking if the amount of water is not too large to pass through the valves and pipes. If they break, they have accomplished their purpose and new ones should be attached.

Eccentric. — Take up lost motion by reducing the brass liners between the lugs on eccentric strap, and unscrew and disconnect the ball joint on the eccentric rod to see that the eccentric strap will turn freely on the eccentric. If a close fit it will heat, cut, seize and break the eccentric rod or valve stem. Allow

the eccentric strap to run loose; no harm if it knocks a little. It will not wear out of round on account of running loose; it is dangerous to run with the strap snug.

Ball joint. — Take up lost motion in the ball joint, on the valve stem, by unscrewing the joint at eccentric rod and turning or filing off the face of the brass part attached to the valve stem, so as to allow the male part to screw in a greater distance.

Connecting rod. — Take up the lost motion on the crank pin bearing by removing the cap and taking out two of the steel liners; take one from each side, put the cap back and set the nuts up snug. Disconnect the cross-head end of the rod by removing cross-head pin, and try lifting the rod up and down to see that it does not pinch the crank pin. If it pinches the pin when the bolts are drawn up snug, place the liners back or substitute thinner ones. Always screw the cap back solid on the liners, and keep in sufficient liners so the cap will not pinch the pin when the bolts are screwed down snug. NEVER RUN THE ENGINE WITHOUT HAVING THE CAP SCREWED UP SOLID AGAINST THE ROD, with liners between if needed, to make the proper fit. When liners are removed be sure to take out an equal amount from each side, because taking out more on one side only is liable to throw the cap at an angle in tightening up the bolts, which, in time, will cause the bolt to break and is liable to wreck the engine.

The brass in the cross-head end of the connecting rod is set up by a wedge. This wedge is drawn down by the steel bolt until the brass is forced solid against the shoulders in the end of the connecting rod, which prevents any movement of the brass. The upper bolt is used to lock the wedge in position; also in withdrawing the wedge when the brass is to be removed.

To take up lost motion in the cross-head end of the connecting rod, remove the brass and file an equal amount, even and square, from each edge of the brass, so as to allow the brass part to come up to the pin. When filing the brass, try the pin in the rod

and do not file enough to allow the brass to pinch the pin when the wedge is screwed *down solid.* If, by mistake, too much is filed off, put in a sheet of copper or sheet brass liner, so the wedge may be drawn snug without pinching the pin.

Cross-head. — For adjusting the lower cross-head slide, take out the cross-pin, turn cross-head $\frac{1}{4}$ round with the lower brass slipper opposite opening in engine frame; loosen nuts and insert paper or thin metal strips between cross-head and slipper. The top slide will never require adjustment. The lower slide should run five years before requiring lining or adjustment. Turn the cross-head pin $\frac{1}{4}$ way around every three months. This will prevent it wearing out of round.

Main bearings. — To take up lost motion in the main shaft bearings, remove the cap and file, scrape or plane an equal amount from each of the babbitt metal liners or strips, which are in the main bearings under the inside edge of the cap. Remove the metal evenly, so the liners will remain of equal thickness at each end. Do not remove enough from the liners to allow the cap to pinch the shaft when the nuts are screwed down snug. If, by mistake, too much metal is removed, put in paper strips on top of the liners so the cap can be screwed down solid without pinching the shaft. Ascertain when the cap pinches the shaft by turning the engine over by hand; it will not turn freely when the cap is too tight. With proper care the main bearings will run two years before requiring adjustment. NONE OF THE BEARINGS OF THE ENGINE SHOULD BE SO TIGHT AS TO PREVENT TURNING THE ENGINE FREELY OVER BY HAND. Always test the engine in this manner after adjusting bearings.

If a bearing heats, stop the engine immediately, take out shaft or box, clean out the cuttings, scrape smooth, clean out oil passages and run bearings loose.

Heating or cutting *will never* occur if liners are put in so caps cannot be set up to pinch the bearings and they receive proper

lubrication with oil free from grit or dirt. After adjusting any of the bearings, run the engine for a few minutes; then stop the engine and feel the bearings which have been adjusted to see if they are running cool. This precaution may obviate having to shut down the engine while performing regular duty.

Do not allow the engine to run with bearings so loose as to thump or pound, as this will cause the bearings to wear out of round. If the shaft or wheels run out of true or wabble, it is because the main bearings are loose and should be taken up. The engine will run smooth and noiseless if bearings are properly adjusted.

THE STEAM CHEST.

Fig. 182 shows a section through cylinder and valve. The steam chest is bored out and fitted with a pair of cylinders or bushings,

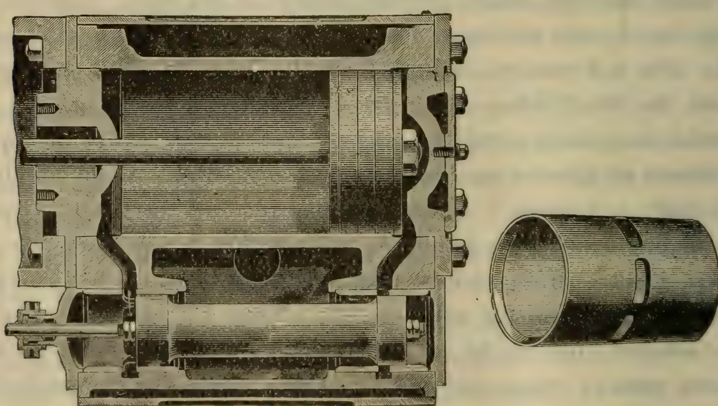


Fig. 182. Cylinder and valve — Ideal engine.

which have supporting bars across the ports, to prevent any possibility of the valve catching upon the ports.

The valve is of the hollow piston type — a hollow tube with a piston at each end. The live steam is entirely upon the outside

of this piston, pressing equally on each end; the exhaust steam is entirely on the inside of the piston, so the valve is perfectly bal-

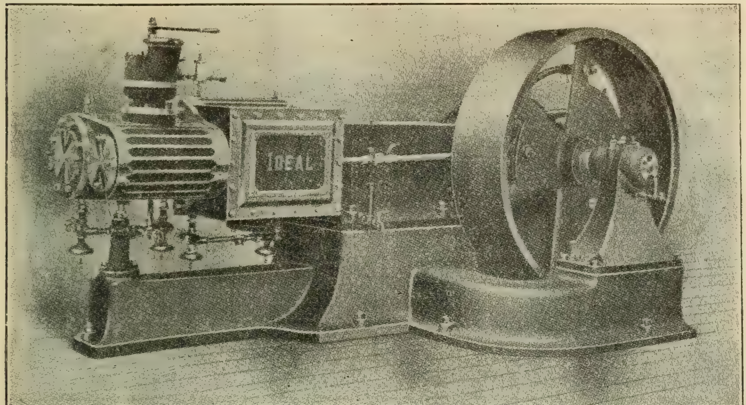


Fig. 183. Tandem compound Ideal engine.

anced and can easily be moved by hand when under full boiler pressure.

Fig. 184 is a cross-section of cylinder and valve of the Tandem Compound engine. The cylinders of the Ideal Compound engine in Fig. 184, the stuffing-box between the two cylinders, is dispensed with entirely. It is replaced by a long sleeve of anti-friction metal. This sleeve is light and free to adjust itself central with the rod. Grooves are turned on the inner surface, so as to form a water packing.

Both valves of engine are controlled by the same governor on the same stem, moving together and varying in stroke as the load and steam pressure vary. This gives the advantage of automatic cut-off in both cylinders and dispenses with the complication of double eccentrics, rock arms, slides and stuffing-boxes.

The high-pressure cylinder has a piston valve, same as used in all Ideal engines. For the low-pressure a flat ported valve is used in order to bring it into line with the high-pressure valve

and keep clearance spaces at minimum, which thus gives a quick and wide opening at the beginning of the stroke, in order to reduce the pressure on exhaust end of high-pressure piston.

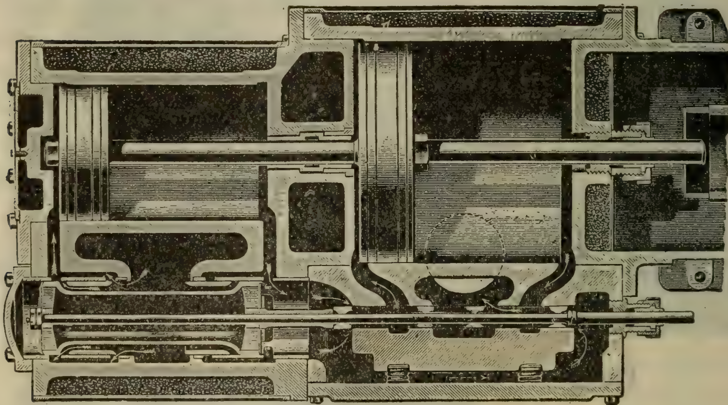


Fig. 184. Section of cylinders of Ideal compound engine.

The cover of this valve is held in place by springs and will lift and prevent excessive pressure in the cylinder from water or other causes.

FOR INDICATING IDEAL ENGINES.

The illustration, Fig. 185, shows the reducing motion attached to engine ready for taking indicator cards.

To apply the Ideal indicator rig: Screw slotted stud in cross-head pin, first removing the cap screw. Set the slot perpendicular to line of motion of cross-head. Set cross-head exactly in center of its travel. Fasten on top of bed where oil funnel is placed, first removing the oil funnel.

Lever should be adjusted so it will travel in slot without strik-

ing bottom, or passing out at top. Make sure that lever will travel freely in slot without binding. Select a hole on string carrier that will give the necessary motion to indicator drum.

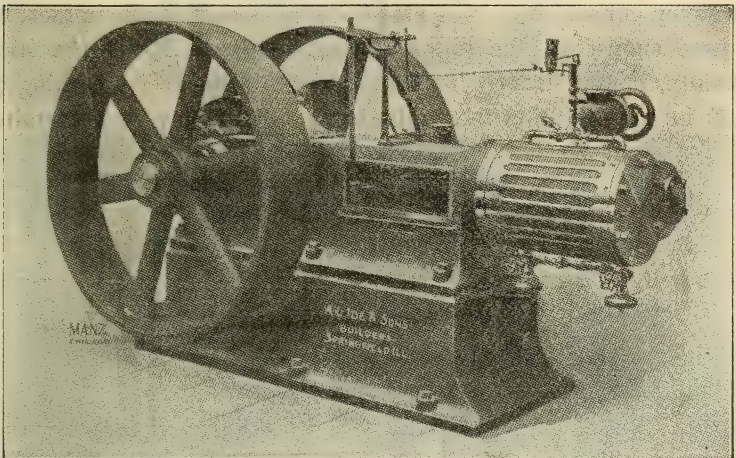


Fig. 185. Method of attaching indicator to Ideal engine.

With string attached from indicator through hole, so adjust this carrier that lines drawn on polished surface shall come exactly parallel with string. Make all adjustments while cross-head is in center of its travel.

HOW TO SET THE VALVE ON A WESTINGHOUSE COMPOUND ENGINE.

The only exact and final setting of the valve is by means of the indicator. As the valves are permanently set and all adjustments made before the engine is shipped, it is not supposed that

the engineer will have occasion to reset them. Should the necessity for setting the valves arise, however, the following method will be sufficiently accurate: Break joints and take off the throttle-valve. The steam ports in the bushing will then be seen through the steam connection *S*. (This opening is on the side in fact, but is here shown on the top for convenience.) Bring the high-pressure piston exactly to the top of its stroke by turning the shaft in the direction the engine runs. This may be ascertained

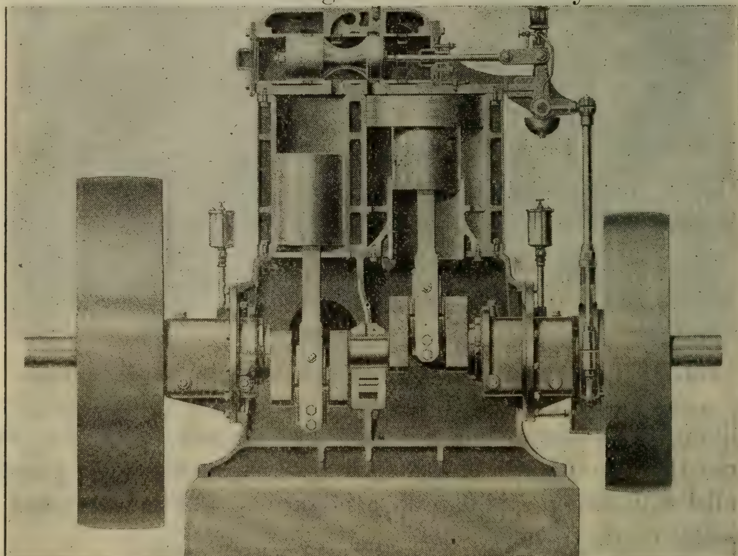


Fig. 186. Westinghouse compound engine.

by either taking off the water relief valve and measuring through its port, or more conveniently, by bringing the middle of the keyway in the shaft exactly over the center of the shaft. The keyways are planed exactly with the cranks, so that the position of the keyway is the position of the high-pressure piston. With this piston at the top of its stroke, the valve edge *a a*, should show about $\frac{1}{16}$ of an inch port or lead, and be moving towards the right when standing behind the engine. If out, it may be brought to position by screwing the valve-stem into or out of the

valve, which is tapped to receive it. Be sure and set the jam nut solid when through.

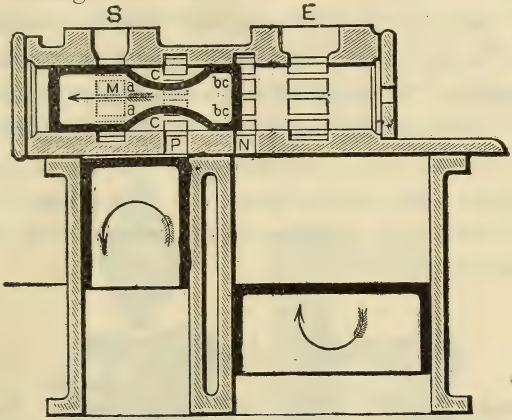


Fig. 187. Cylinders of Westinghouse compound.

SOME POINTS ON CYLINDER LUBRICATION.

In the first place, use the best automatic feed cup that can be secured. Don't be satisfied with the old-fashioned direct feed, or a cheap automatic. A good cup will save many a hundred per cent on its cost in a year. Don't get the kind which, on account of its peculiarity of feed, is adapted for a light oil only; it will then be shut out from using a dark oil, which may be far more serviceable and economical in every respect. Get a cup where the drop of oil cuts off square and passes either down or up through a glass tube into the steam pipe. This kind will feed oil perfectly; it is well to use this kind.

Take good care of the cup. Don't let it leak around the glass tubes or other joints, for if it does the water will escape as it condenses, and the oil will clog up the escape pipe and stop feeding. Use in it only the best grades of cylinder oil, made by large manufacturers of established reputation. Don't run in the cylinders any kind of poor stuff that may be offered, because it is cheap; it is a dangerous experiment. Feed a good

oil sparingly — don't drench the cylinder. Too much oil is as bad as water in the cylinder. Engineers have been known to run a couple of quarts per day of cheap oil into an ordinary sized cylinder, and thought they were doing just right; this is positive abuse of an engine. In almost all cases where too much oil is fed,— cut it down. Two to four drops per minute on engines from 50 to 150 H. P. are all that is necessary, if the oil is good. Just enough to do the work and no more, will afford best results. As long as the valve stem does not cause trouble, rest assured the valves are working smoothly.

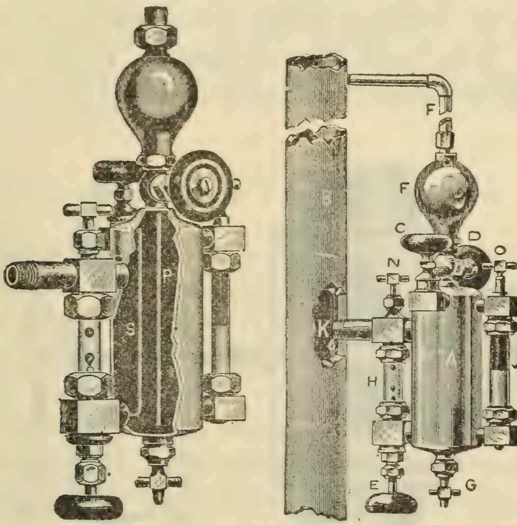
AUTOMATIC LUBRICATORS.

An automatic sight feed lubricator should be furnished with every engine which enables the engineer to see the oil as it is fed drop by drop to the engine. The construction of these lubricators is such that the steam entering a chamber is condensed and this water of condensation finds its way into another compartment of the lubricator, wherein is contained the oil to be fed to the engine. The drop of water, by reason of its greater specific gravity, seeks the bottom of this oil compartment and forces out an equivalent bulk of oil into the steam pipe, whence it is carried by the current of steam into the cylinders and is distributed upon the wearing surfaces intended to be lubricated. This method insures regularity and economy.

There are numerous automatic lubricators made by various manufacturers throughout the country, many of which will perform their functions successfully. The illustrations on the opposite page represent what may be called the standard type of hydrostatic lubricator employed for lubricating the valves and pistons of steam engines.

This is the up-feed cup, showing an external view and sectional view of the same. Attachment is made to the steam-pipes

at the points *F* and *K*. In operation, the condensing chamber *F* provides for the condensation of steam, which enters at the pipe *F*. This water of condensation passes down through the valve *D* and through the tube *P* shown in the section and discharges into the bottom of the oil vessel *A*. This vessel is filled with oil when the cup is started, the height of oil being shown in the index glass *J*.



Figs. 188 and 189. Sight feed cylinder lubricator.

The operation is as follows: The valve *N* being opened, the valve *D* is opened and the drop of water is allowed to pass from the condensing chamber *F* downward through the water tube and into the bottom of the oil chamber *A*, where it displaces a drop of oil of equal bulk on account of its greater gravity, and this drop of oil is forced out past the valve *E*, making its appearance in the feed glass *H*, as it starts on its way to the steam-pipe. It is carried by the current of steam to the engine and lubricates the valve and the pistons. When the oil cup is empty, the valve *D*

is closed and the drain valve *G* is opened, which will allow the water in the oil chamber to be blown out preparatory to the refilling at the plug *C*. By opening the valves *G* and *D*, steam will be blown through the sight glass *J*, thereby clearing the same from any clogging up of the oil, which would disfigure it. The amount of oil to be fed by the lubricator will be regulated by the valve *D*, controlling the amount of water admitted, and the valve *E* controlling the discharge of the oil into the sight glass. The valve *N* is to be left wide open in operation and its object is to provide for the accidental breaking of the glass *H*.

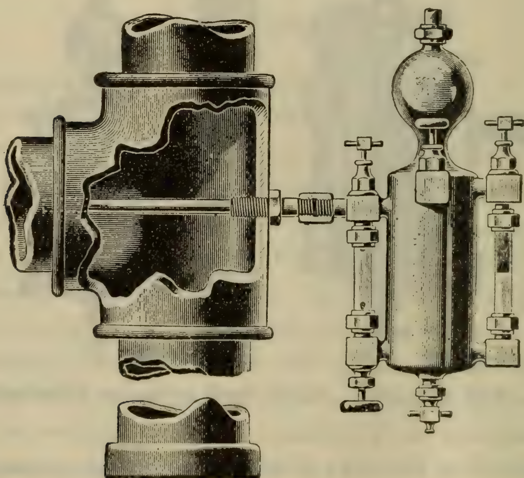


Fig. 190. Proper method of attaching cup to prevent the oil from dropping into the well, and not going into the cylinder.

These cups should be attached to the steam pipe, in strict accordance with the instructions contained in the box in which the lubricator is packed. The greatest enemy to proper performance is leakiness; all joints must be absolutely tight, otherwise the

water of condensation, instead of performing its duty of displacing the oil, will ooze out at the leaks and the cup will refuse to work. In most cases, provision is made for a column of water, which may stand 12″ or more in height and enable the cup to work more positively, by giving it a greater pressure in the displacement chamber, due to the height of the column. A suitable oil is essential to the proper working of such a lubricator, as well as to the proper lubricating of a steam engine. An improper oil will not feed through the cup as it should, on account of its disposition to disintegrate and go off in bubbles, when exposed to the heat of the steam.

SETTING A PLAIN SLIDE VALVE WITH LINK MOTION.

The setting of a slide valve operated by a link motion does not differ materially in principle from the method pursued when setting the ordinary slide valve driven by one eccentric. A link motion may be considered as a means of driving a valve by two independent eccentrics, either of which controls the functions of the valve wholly or in part, according to the position of the link. Thus when the link is in either extreme position, the eccentric driving that end of the link in line with the link-block pin may be considered as being entirely in control of the valve action, and, *vice versa*, when the link occupies the other extreme position of its throw, as actuated by the reverse lever, the other eccentric becomes possessed of the controlling function. Practically, however, the operation of the link motion is very complicated and the movement of one eccentric materially modifies the action of the other. Since the interfering action is least at the extreme positions of the link and greatest in mid-gear, the plan is followed of setting the valve with the link in full gear both forward and backward motion, and, as before stated, the procedure is on the theory of independent action of the eccentrics.

In the accompanying diagram, a link motion is shown driving a plain slide valve without the intervention of a rocker. Each eccentric is set with reference to the crank pin, the same as it would be with a simple slide-valve engine. The eccentric *A* is set on the shaft with the same angular advance, *QMO*, as would be required for an ordinary engine to run in the direction indicated by the arrow. Now, since the crank pin is at *C*, if it were necessary to reverse the simple engine with one eccentric, it would be necessary to change the position of the eccentric so that instead of being ahead of the bottom quarter line *QM*, it would be ahead

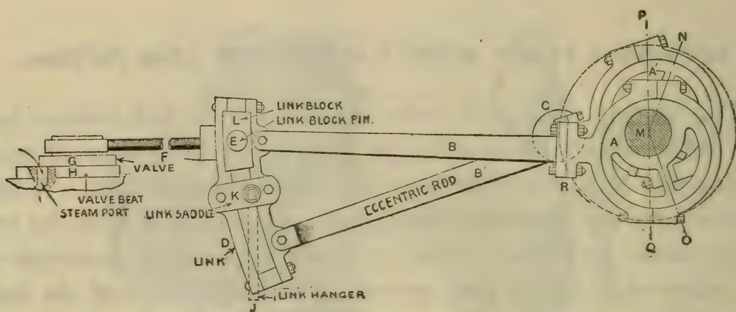


Fig. 191. Diagram of link reversing gear.

of the top quarter line *PM* by an amount of angular advance made necessary by the lap and lead of the valve. Therefore, the eccentric would come in the position of the eccentric A^1, or with its center line coinciding with *MN*, giving it the angular advance *PMN*. Now it should be clear that if an engine is to be equipped with two eccentrics, so that it may run with equal facility in either direction, they will occupy the positions *A* and A^1. We will suppose that an engine having a link motion is to be overhauled and the valve motion to be properly set. This will mean that the eccentrics will be properly located for the correct angular advance, and that the eccentric rods will be adjusted to the right length. When these conditions are obtained, the valve should

perform its functions properly in both forward and backward motions, and also when the link is "hooked up."

Before starting to set the valve, it is best to take a general survey of the valve motion parts and see if the eccentrics are somewhere near the proper location on the shaft relative to the crank-pin. If they are obviously much out of position, they should be shifted and adjusted as near the correct position as possible by the eye; doing this at the beginning will often save confusion and much time. The dead centers will be found by the method given on page 195. The operation should be carefully performed, as upon it depends the success of the work. After having found the dead centers and having them marked so that no mistake will occur when "catching" them with the tram, the valve positions may be taken for the four positions, that is, front and back centers in forward motion, and the front and back centers in backward motion. Put the reverse lever in full gear in one motion or the other, whichever is most convenient, and turn the fly-wheel in the direction the engine would run for the given reverse lever position. Suppose the link stands in the position shown in the diagram, the fly-wheel should be turned in the direction indicated by the arrow until the dead center is reached, which is known when the tram drops into the prick mark. The position of the valve is then noted and a measurement taken. If the valve shows the steam port open, measure the distance with a steel scale, or it may be done by sharpening a stick wedge-shaped and shoving it into the opening. By noting the depth to which it goes at the valve face the opening can be readily measured on the removal of the wedge. We will suppose the distance is found to be $\frac{3}{8}''$. The measurement should be set on a sheet of paper laid out as follows: —

FORWARD MOTION.	BACKWARD MOTION.
Front center,	Front center,
Back center.	Back center, $\frac{3}{8}''$ lead.

It will be seen that the valve opening is set down as being $\frac{3}{8}''$ lead, and as being on the back center in the backward motion. After having verified the measurement taken, the engine can be "turned over" in the same direction as before until the opposite dead center is caught by the tram. It may be found that the valve does not show open in this position but covers the steam port. To find the position of the valve edge relative to the steam port, scribe a line in the valve seat face along the edge of the valve and then turn the fly-wheel until the valve uncovers the steam port. The distance the valve laps over when the crank is on this dead center can then be readily measured. Suppose the distance is found to be $\frac{1}{8}''$. It is set down on the log as follows: —

FORWARD MOTION.	BACKWARD MOTION.
Front center.	Front center, $\frac{1}{8}''$ blind.
Back center.	Back center, $\frac{3}{8}''$ lead.

The valve position is put down as being $\frac{1}{8}''$ blind, which is the same as saying that it has $\frac{1}{8}''$ negative lead, and is fully as comprehensive as the latter term. The reverse lever should now be thrown into the opposite gear and the measurements taken for both front and back centers the same as has been described for the backward motion. It may now be supposed that when all the measurements have been taken the log reads as follows: —

FORWARD MOTION.	BACKWARD MOTION.
Front center, $\frac{1}{4}''$ blind.	Front center, $\frac{1}{8}''$ blind.
Back center, $\frac{5}{16}''$ lead.	Back center, $\frac{3}{8}''$ lead.

When in forward motion, the valve is open $\frac{5}{16}''$ on the back center and lacks $\frac{1}{4}''$ of being open when the crank is on the front center. The total lead due to the angular position of the eccentric is $\frac{5}{16}''$ minus $\frac{1}{4}'' = \frac{1}{16}''$. One-half the total lead should be given to each edge of the valve so that it will be necessary to lengthen the eccentric rod B^1, $\frac{1}{32}'' + \frac{1}{4}'' = \frac{9}{32}''$ to get the valve

into its proper position. A little reflection will show the reason for lengthening the eccentric rod B^1. In speaking of the front and back centers, they are taken to coincide with the crank and head ends of the cylinder. When the piston is at the crank end of the cylinder, the crank is on the front center. By referring to the log it will be seen that to adjust the backward eccentric rod B, it will also be necessary to lengthen it. The valve is $\frac{1}{8}''$ blind on the front center and has $\frac{3}{8}''$ lead on the back center. The total lead is, therefore, $\frac{3}{8}''$ minus $\frac{1}{8}'' = \frac{1}{4}''$. One-half $\frac{1}{4}'' = \frac{1}{8}''$, which being added to the amount the valve is lapped on the front center, makes $\frac{1}{4}''$, or the amount the eccentric rod B will have to be lengthened to make the valve open equally at each end of the piston stroke. The opening the valve has when the crank is on the centers is called the lead and in the case of the backward motion, it is found that after the eccentric rod is lengthened, the lead is $\frac{1}{8}''$, which is too much for most cases and in this one we can assume that $\frac{1}{32}''$ would be about right.

Before explaining the adjustment of the eccentric for the correct angular advance, it will be in order to call attention to the necessity of making the adjustment for the eccentric rod lengths first. The eccentric rods are lengthened or shortened, as the case may require, by inserting or removing liners between the eccentric rods and straps at R. Other forms of construction provide different means for adjustment, but the principle is the same in each. It will be noted that the correct length for the two motions is obtained by adjusting the eccentric rod corresponding to that motion. Any attempt to correct an irregularity by changing the length of the valve rod F will result erroneously, unless both eccentric rods require the same amount of movement and in the same direction. After having adjusted the eccentric rods to the correct lengths, the angular advance of the eccentric A can be changed. Place the crank on a dead center and have the reverse lever thrown in the backward motion and then

loosen the set screws that hold the eccentric to the shaft and turn it towards the crank until the valve shows open $\frac{1}{32}''$, and then tighten the set screws on the shaft. After all the adjustments have been effected, it is always advisable to turn the engine over again and catch all the dead centers, so that the correctness of the adjustments can be verified. After taking the new log, it will usually be found that some slight irregularities have been introduced, especially if any of the adjustments have been considerable, as the changes made for one motion will affect the other slightly.

The link motion shown in the cut is so connected that the lead increases as the link is shifted towards the center. If the eccentric rods be oppositely connected to the link, the engine will run in an opposite direction for a given reverse lever position and the lead will decrease as the lever is shifted towards the center. The link motion for hoisting engines is quite commonly connected in this manner, for the reason that the engine will stop when the lever is put on the center, which is not the case when connected as shown. Of course, in such a case, the admission and cut-off take place at the same position in the stroke and the compression is high, but with a light load the engine will run on the center, which is considered objectionable in the case of the hoisting engine.

VALVE-SETTING FOR ENGINEERS.

Plain slide-valve. — The plain slide-valve, while the simplest valve made, is perplexing to one who has not made a study of it. Unless one understands the principles of the valve and its connections, he will probably meet with trouble when he attempts to set it. We will first place the engine (see p. 195) on the dead center, and will simply explain the other steps that have to be taken. In the first place, it should be understood what result is obtained by adjusting the position of the eccentric

and the length of the valve stem. The position of the eccentric, when the valve is set, depends upon which way the engine is to run and whether the valve is connected directly to the eccentric or whether it receives its motion through a rocker which reverses the motion of the eccentric. When the valve is direct connected, the eccentric will be ahead of the crank by an amount equal to 90°, plus a small angle called the angular advance. When a reversing rocker is used, the eccentric will be diametrically opposite this position, or it will have to be moved around 180° and will follow instead of lead the crank. Shifting the eccentric ahead has the effect of making all the events of the stroke come earlier, and moving it backwards has the effect of retarding all the events. Lengthening or shortening the valve stem cannot hasten or retard the action of the valve, and its only effect is to make the lead or cut-off, as the case may be, greater on one end than on the other. The general practice is to set a slide-valve so that it will have equal lead. The lead is the amount that the valve is open when the engine is on the center. To set the valve, therefore, put the engine on the center, remove the steam-chest cover so as to bring the valve into view, and adjust the eccentric to about the right position to make the engine turn in the direction desired. Now make the length of the valve-spindle such that the valve will have the requisite amount of lead, say $\frac{1}{16}$ of an inch, the amount, however, depending upon the size and speed of the engine. Turn the engine over to the other center and measure the lead at the end. If the lead does not measure the same as before, correct half the difference by changing the length of the valve-stem, and half by shifting the eccentric. Suppose, for example, that the lead proved to be too great on the head end by half an inch. Lengthening the valve-stem by half of this, or $\frac{1}{4}$ inch, would still leave the lead $\frac{1}{4}$ inch too much on the crank end. That is to say, the valve would then open too soon at both head and crank ends, and to correct this, the eccentric would

have to be moved back far enough to take up the other quarter-inch. Sometimes it is not convenient to turn the engine over by hand, in which case the valve may be set for equal lead as follows: To obtain the correct length of the valve-stem, loosen the eccentric and turn it into each extreme position, measuring the total amount that the valve is open to the steam ports in each case. Make the port opening equal for each end by changing the length of the valve-stem. This process will make the valve-stem length as it should be. Now put the engine on a center and move the eccentric around until the valve has the correct lead and fasten the eccentric in that position. This will determine the angular advance of the eccentric.

The plain slide valve. — The function of the slide-valve is to admit steam to the piston at such times when its force can be usefully expended in propelling it, and to release it when its pressure in the cylinder is no longer required. Notwithstanding its extreme simplicity as a piece of mechanism, no part of the engine is more puzzling to the average engineer when the problem to be solved is to determine beforehand the results which will be produced by a given construction and adjustment, or the proportions and adjustment required to produce given results. All who have had any experience in constructing and setting slide-valves are aware, in a general way, that the events of the stroke cannot be independently adjusted; for instance, a cut-off earlier than about $\frac{5}{8}$ of the stroke.

To set a slide valve. — The valve should be set in such a manner that when the engine is on the dead center, the part admitting the steam to the cylinder is open a small amount as shown in Fig. 185, which is called lead. The object of lead is to enable the steam to act as a cushion against the piston before it arrives at the end of the stroke, to cause it to reverse its motion easily, and also to supply steam of full pressure to the piston the instant it has passed dead center. The lead required varies in different engines from

$\frac{1}{64}$ to $\frac{3}{16}$ without regard to size or kind. Fig. 192 also shows the position of eccentric, which should always be set ahead of the

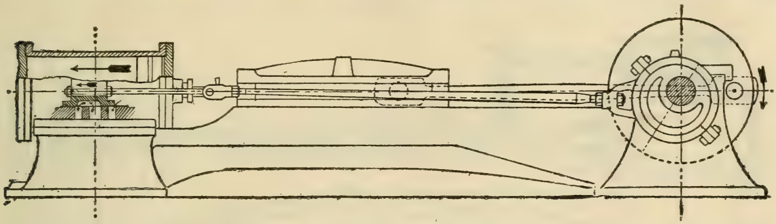


Fig. 192. At point of taking steam.

crank at an angle of 90°, plus another angle, called the "angular advance." When the valve is to have lead the angular advance must be a little greater than when no lead is desired.

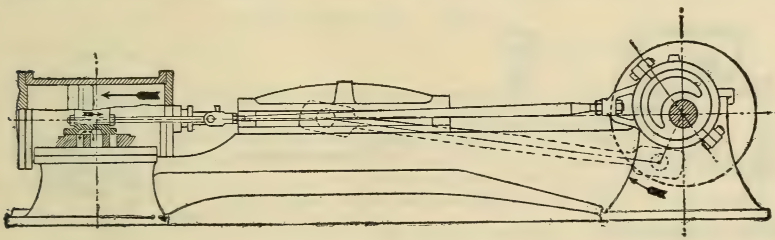


Fig. 193. At point of cut-off.

Fig. 193 shows the position of eccentric at point of cut-off; also position of piston.

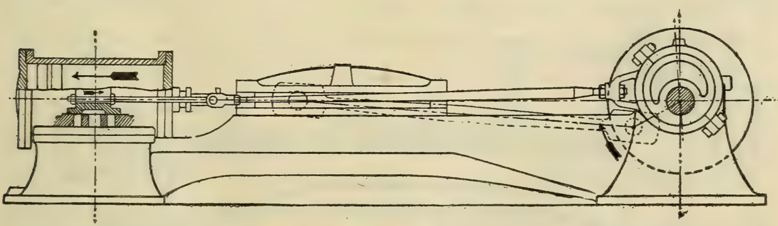


Fig. 194. Position when compression begins.

Fig. 194 shows position of valve when compression begins. It also shows position of eccentric. The compression at the left

end, towards which the piston is moving, has just commenced, and the exhaust is about to take place from the other end.

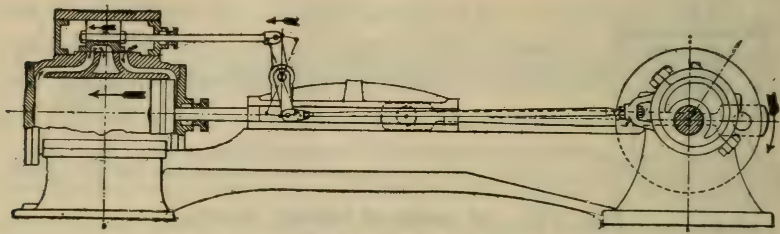


Fig. 195. At point of taking steam.

Fig. 195 shows the position of eccentric and valve in an engine with a rocker-arm.

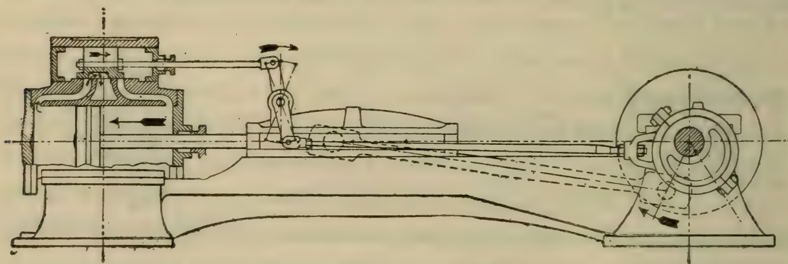


Fig. 196. At point of cut-off.

Fig. 196 shows the position of valve and eccentric at point of cut-off.

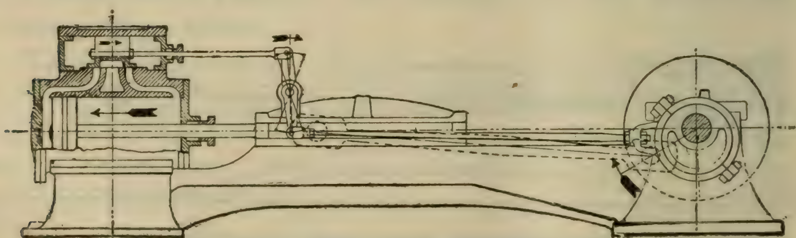


Fig. 197. Showing point of compression.

CHAPTER XIII.

TAKING CHARGE OF A STEAM POWER PLANT.

It is frequently the case that an engineer, on assuming charge of a steam power plant, proceeds as though he were thoroughly familiar with the condition of the engine, boiler and entire surroundings. He plunges headlong into his duties, without first taking his bearings. A skillful physician on taking a case, would not proceed in this manner ; neither would a lawyer. The physician would feel the patient's pulse, look at his tongue, take his temperature, observe his color and ask a number of questions, all for the purpose of enabling him to make a correct diagnosis of the patient's ailment. The first duty of an engineer, when he takes charge of a plant, is to ascertain the arrangement and condition of the plant. Since the boiler is the most important member of the plant, it should be the first to engross his attention, and it, together with its connections, should be examined as closely as time and surrounding conditions will permit. He should look the boiler all over, internally and externally, if possible, in search of

mud, scale, grooving, pitting and defective braces. The furnace should be examined next, in view of burnt-out brickwork, grate bars and door linings. It may be that the furnace has distorted or cramped proportions, or it may be too large. The bridge-wall may be so constructed as to huddle the flames in one spot on the fire sheets of the boiler; or it may be of such shape and in such condition as to cause the ignited gases to become dissipated in the combustion chamber. Even the combustion chamber itself may require the service of a bricklayer. He should next examine the safety valve and see that it is of ample capacity to relieve the boiler of surplus steam, and that it is in thorough working order. The first duty of an engineer when entering his plant at any time, is to ascertain how the water in the boiler stands, or, in other words, just how much water the boiler contains. He should open the gauge cocks first and note what comes from each in turn; then open the cocks or valves connecting the glass gauge and note the water line there shown. He should also blow the water column out, in case any sediment may have choked any of the passages, which would be liable to give a false impression as to the actual quantity of water contained in the boiler. Should the water be found at the correct height, he may now proceed to get up steam; open the damper, pull down the banked fire and spread it evenly over the grate, adding a quantity of green fuel. Allow the steam to rise slowly; do not force it. This applies especially to raising steam in a boiler which has been cold, as the expansion of the parts of the boiler due to the heat should take place slowly and evenly; otherwise, the life of the boiler will be shortened. While waiting for the steam to come up to the desired point, the engineer should now get his engine ready for the day's run. Fill all the oil cups and cylinder lubricator, so as to be ready to operate as the engine starts. With a hand oil squirt can, go around all the small brasses, connections, etc., and, in a word, well lubricate all the parts where friction takes place. If

he uses an oil pump for the cylinder and valves, it would be well to inject a small quantity of cylinder oil before the engine is started, while the stop-valve is open, during the time the engine is being "warmed up." After the engine cylinder is warmed through, the fire should again be looked at, and dealt with according to the indications. Of course, the water gauge glass must be looked at frequently, not only while raising steam in the morning, but at all times while the boiler is in operation.

Everything being in readiness, the engine is started slowly at first, the speed being gradually increased until the limit is reached. The day's run is now fairly commenced. A boiler should be blown down one gauge every morning before starting the day's run to get rid of the mud, scale or anything that is held in mechanical suspension in the water. Before starting in the morning and at noon is the best time to do this, as the sediment has settled to the bottom during the night, after the circulation of the water has stopped. When blowing a boiler down, always remember to open the blow-valve slowly — be careful not to blow too long, and then to close the valve slowly.

An engineer or attendant cannot be too careful in handling the many appliances with which a steam plant is equipped. The principal things to which an engineer should give his attention during the operation of his boiler day by day are, as follows: The maintenance of the water at the proper level, as near as possible, and avoiding fluctuations in the pressure of steam. See that the firing is done correctly and economically so as to obtain from every pound of coal all that is possible under the conditions existing. The raising of the safety valve from its seat, at least once daily; the blowing out of the water column twice daily, or oftener, if the water used is very dirty; the frequent opening of the water gauge cocks, or try cocks, as they are sometimes called, and not depending entirely on the gauge glass for the correct height of water; the blowing down of the boiler

one gauge every day; the keeping of all valves, cocks, fittings, steam and water-tight, clean and in good working order.

When shutting down the plant for the night, the fires should be cleaned out and the live coals shoved back on the grates and banked; that is, green coal should be thrown upon them, sufficiently thick to cover all the glowing fuel. Pump in the water until it reaches the top of the glass gauge. This should be done to insure a sufficient quantity from which to blow down in the morning, and also to allow for any small leaks. Then close the cocks or valves connecting the glass gauge. Should this glass break during the night and the valves be left open, there would not be much water to start with in the morning. Leave the damper open a little, just sufficient to allow the gases which will rise from the banked fires to escape up the chimney. Finally, make sure that all the valves about the plant which should be closed, are closed; and all those which should be left open, are open. Of course, the foregoing is applicable to a plant where there is no night engineer. But in any case, no matter how many assistants an engineer may have under his control, he should be familiar with all details of the plant under his charge.

One of the most important points in connection with the operation of a steam boiler, is the preventing of corrosion, both internally and externally. One of the best aids to secure the well working and longevity of the steam boiler, or, in fact, the whole plant, is by being regular and punctual in a certain course of treatment, which has been proven to be effectual and beneficial in its results. All conditions do not require the same methods of treatment; therefore, it is absolutely necessary that the engineer in charge familiarize himself with all the conditions under which his plant is running, for then, and then only, can he intelligently prescribe and act accordingly. Above all, let him remember the adage, "Eternal vigilance is the price of safety," especially where a steam boiler is concerned.

ECONOMY IN STEAM PLANTS.

In these days of close figuring upon expense in office buildings and manufacturing plants, what may at first appear insignificant items may actually make all the difference between a good margin of profit and an actual loss.

The fuel expense is one of the largest in the operation of the majority of plants, and any reduction which can be made in the amount of fuel used, while maintaining the same amount of power, is considered a direct gain. The evaporation of more than nine pounds of water per pound of coal, is looked upon with suspicion by many, as it is not thought possible to obtain more than this amount in even the best designed and well regulated furnaces and boilers, especially when the firing is done by hand. The actual value of the fuel depends upon the way in which it is used, fully as much as on any other factor. The heat unit in the coal should be as much as possible utilized, as in one pound of good steam coal there is about 14,000 B. T. U., and about 10,000 of this amount can be utilized, so that 4,000 heat units are lost. The mixture of gases in a furnace depends upon the amount of air used. One pound of coal requires, theoretically, about twelve pounds of air to burn completely. But, in practice, about twice this amount is required in the present boiler furnace. To have good combustion coal requires a good draft. The gases are consumed near the fire, and the waste gases carry the heat to the boiler on their way to the stack. The boiler ought to have sufficient heating surface, or the hot wasted gases ought to travel a sufficient distance to be cooled down to about 350 degrees Fahrenheit; which temperature is found high enough to produce a good draft in a stack of, at least, 100 feet high.

How a bad draft will unnecessarily increase the coal bill, is this: That of all the fuel burnt to perform certain work, a certain proportion is consumed to keep the heat of the furnace up

to say, 212 degrees Fahr., without making any steam whatever which is available for work. This quantity varies from 20 to 30 per cent, according to conditions, which are affected by various causes, such as leakages of steam, air, or water. Now, the only available power for work which we get from our fuel is the margin between this, say thirty per cent required for the said purpose, and what we generate above that. An engineer should notice the general condition of his boiler or boilers, and the equipments of same; he should examine the boiler both inside and outside, ascertain the dimension of grates, heating surfaces, and all important parts. The area of heating surfaces is to be computed from the outside diameter of water-tubes, and the inside diameter of fire-tubes. All the surfaces below the main water level which have water on one side and products of combustion on the other, are to be considered as water-heating surfaces. If he finds that the boiler does not come up to what he thinks it should, he should put the boiler and all its appurtenances in first-class condition. Clean the heating surfaces inside and outside of boiler, remove all scale from flues and inside of boiler; remove all soot from inside of flues, all ashes from the flame-bed or combustion chamber, and all ashes from smoke connections. Close all air leaks in the masonry and poorly fitted cleaning door. See that the damper in britching or smoking-flue will open wide and close tight. Test for air leaks through the crevices, by passing the flame of a candle over cracks in the brick work. A good, attentive fireman, who understands his business and will keep his bars properly covered without choking his fires, is really worth double the wages of an ignorant or inattentive one, as his coal bills would certainly prove. All an engineer can do is to keep the steam piston and valve or valves tight. Also the drains from his engine, and all drains on steam traps in the plant tight; also, his engine cleaned and well-oiled, and not keyed up too tight. If in a heating plant, he should see that the back pressure valve is

at all times tight, as it does not take much of a leak to show a difference in his coal bill at the end of a month. He should keep all valves in the pumps in his plant tight, and see that the pump piston is packed, but not too tight. After a pump is packed, he should be able to move it back and forth by hand; if the pump valves leak he can take them out and smooth them up with sand-paper. He should see that the feed-water to the boiler is at least 208 degrees Fahrenheit; if it is under 204 degrees, his heater is not right, as the poorest heater will heat the feed-water to 204; it would be well to overhaul the heater — it may be full of scale; or, if an open heater, the spray may be off. In most first-class plants, the feed-water is 212 Fahrenheit.

PRIMING.

The term priming is understood by engineers to mean the passage of water from the boiler to the steam cylinder, in the shape of spray, instead of vapor. It may go on unseen, but it is generally made manifest by the white appearance of the steam as it issues from the exhaust-pipe as moist steam, which has a white appearance and descends in the shape of mist, while dry steam has a bluish color and floats away in the atmosphere. Priming also makes itself known by a clicking in the cylinder, which is caused by the piston striking the water against the cylinder head at each end of the stroke. Priming is generally induced by a want of sufficient steam-room in the boiler, the water being carried too high, or the steam-pipe being too small for the cylinder, which would cause the steam in the boiler to rush out so rapidly that, every time the valve opened, it would induce a disturbance and cause the water to rush over into the cylinder with the steam.

TABLE OF PROPERTIES OF SATURATED STEAM.

Pressure in pounds per square inch above vacuum	Temperature in degrees Fahrenheit.	Total heat in heat units from water at 32°.	Heat in liquid from 32° in units.	Heat of vaporization or latent heat in heat units.	Density or weight of a cubic foot in pounds.	Volume of one pound in cubic feet.	Factor of equivalent evaporation at 212°.	Total pressure above vacuum.
1	101.99	1113.1	70.0	1043.0	0 00299	334.5	.9661	1
2	126.27	1120 5	94.4	1026.1	0.00576	173 6	.9738	2
3	141.62	1125.1	109.8	1015.3	0.00844	118.5	.9786	3
4	153.09	1128.6	121.4	1007.2	0.01107	90 33	.9822	4
5	162.34	1131.5	130.7	1000.8	0.01366	73 21	.9852	5
6	170.14	1133.8	138 6	995.2	0.01622	61 65	.9876	6
7	176 90	1135.9	145.4	990.5	0.01874	53.39	.9897	7
8	182.92	1137.7	151.5	986 2	0.02125	47.06	.9916	8
9	188.33	1139.4	156.9	982.5	0.02374	42 12	.9934	9
10	193.25	1140.9	161.9	979 0	0.02621	38.15	.9949	10
15	213 03	1146.9	181 8	965.1	0.03826	26.14	1.0003	15
20	227 95	1151.5	196.9	954 6	0.05023	19.91	1.0051	20
25	240.04	1155.1	209.1	946.0	0.06199	16.13	1 0099	25
30	250.27	1158.3	219 4	938.9	0 07360	13.59	1.0129	30
35	259.19	1161.0	228.4	932.6	0 08508	11.75	1.0157	35
40	267.13	1163.4	236.4	927.0	0.09644	10.37	1.0182	40
45	274.29	1165.6	243.6	922.0	0.1077	9.285	1.0205	45
50	280.85	1167.6	250.2	917 4	0 1188	8.418	1.0225	50
55	286 89	1169.4	256.3	913.1	0.1299	7 698	1.0245	55
60	292.51	1171.2	261 9	909.3	0 1409	7.097	1.0263	60
65	297.77	1172.7	267.2	905.5	0.1519	6.583	1.0280	65
70	302 71	1174.3	272.2	902.1	0.1628	6.143	1.0295	70
75	307 38	1175.7	276.9	898 8	0.1736	5.760	1 0309	75
80	311 80	1177.0	281.4	895.6	0.1843	5.436	1.0323	80
85	316.02	1178 3	285.8	892.5	0.1951	5.126	1 0337	85
90	320.04	1179.6	290 0	889.6	0.2058	4.859	1.0350	90
95	323.89	1180.7	294.0	886.7	0.2165	4.619	1.0362	95
100	327.58	1181.9	297.9	884.0	0.2271	4.403	1.0374	100
105	331 13	1182.9	301 6	881.3	0 2378	4.205	1.0385	105
110	334.56	1184 0	305 2	878 8	0.2484	4.026	1.0396	110
115	337.86	1185.0	308.7	876 3	0.2589	3.862	1.0406	115
120	341.05	1186.0	312.0	874.0	0.2695	3.711	1 0416	120
125	344.13	1186 9	315.2	871.7	0.2800	3.571	1.0426	125
130	347.12	1187 8	318.4	869.4	0.2904	3.444	1.0435	130
140	352.85	1189.5	324.4	865.1	0.3113	3 212	1 0453	140
150	358.26	1191.2	330.0	861.2	0.3321	3.011	1.0470	150
160	363.40	1192.8	335.4	857.4	0 3530	2 833	1.0486	160
170	368 29	1194.3	340.5	853 8	0.3737	2.676	1.0502	170
180	372.97	1195.7	345.4	850.3	0.3945	2.535	1.0517	180
190	377.44	1197.1	350.1	847.0	0.4153	2.408	1.0531	190
200	381.73	1198.4	354.6	843 8	0.4359	2.294	1 0545	200
225	391.79	1201.4	365.1	836.3	0.4876	2.051	1 0576	225
250	400.99	1204.2	374.7	829.5	0.5393	1.854	1.0605	250
275	409.50	1206.8	383.6	823.2	0.5913	1.691	1 0632	275
300	417.42	1209.3	391 9	817 4	0.644	1 553	1.0657	300
325	424.82	1211.5	399.6	811 9	0.696	1 437	1.0680	325
350	431 90	1213.7	406.9	806.8	0.748	1.337	1 0703	350
375	438 40	1215.7	414.2	801 5	0 800	1.250	1.0724	375
400	445.15	1217.7	421.4	796 3	0.853	1.172	1.0745	400
500	466.57	1224.2	444.3	779.9	1.065	.939	1.0812	500

The gauge pressure is about 15 pounds (14.7) less than the total pressure, so that in using this table, 15 must be added to the pressure as given by the steam gauge. To ascertain the equivalent evaporation at any pressure, multiply the given evaporation by the factor of its pressure, and divide the product by the factor of the desired pressure. Each degree of difference in temperature of feed-water makes a difference of .00104 in the amount of evaporation. Hence, to ascertain the equivalent evaporation from any other temperature of feed than 212°, add to the factor given as many times .00104 as the temperature of feed-water is degrees below 212°. For other pressures than those given in the table, it will be practically correct to take the proportion of the difference between the nearest pressures given in the table. Example: If a boiler evaporates 3000 lbs. of water per hour from feed-water at 200 degs. Fah. into steam at 100 lbs. per sqr. in. by the gauge, what is the equivalent evaporation "from and at" 212°? Ans. 3159.24 lbs.

Operation: Temperature of feed-water = 200 degs.

Then, 212 — 200 = 12 = difference in temperature.

Then, 15 added to the gauge pressure = 115.

Looking in the above table we find the factor 1.0406.

Then, .00104 × 12 = .01248.

And,
$$\begin{array}{r} 1.0406 \\ .01248 \\ \hline 1.05308 \end{array}$$

Then, 3000 × 1.05308 = 3159.24 lbs. the equivalent evaporation.

The H. P. of this boiler would be 91.57.

Thus, $\frac{3159.24}{34.5} = 91.57.$

HIGH PRESSURE STEAM.

It is generally believed that high-pressure steam is cheaper to use and costs but little more to generate than low pressure steam. A study of a table of the properties of saturated steam, to be found on another page in this book, will show why high-pressure steam is economical to generate, and a few calculations will prove instructive by showing what may be excepted from its use. To generate one pound of steam at 25 lbs. pressure, absolute, requires an expenditure of 1,155 thermal units, and to generate steam at 200 lbs. pressure, absolute, requires 1,198 thermal units, or an increase of only 43 thermal units for an increase of 175 lbs. pressure. Further investigation shows that the temperature of steam at 25 lbs. pressure is 240° and at 200 lbs. pressure, 382°, the difference, 142, being the number of degrees that the temperature of steam is raised with an expenditure of 43 thermal units. To put it in another way, the temperature of the steam has been raised nearly 60 per cent, with an increase of less than 4 per cent in the number of thermal units. It is convenient to consider that the generation of steam takes place by two different steps, one of which is raising the water from 32° to the temperature corresponding to the pressure of the steam, and the other is giving off the steam at this pressure, which process absorbs a quantity of heat that becomes latent or non-sensible. At 25 lbs. pressure, the sensible heat required to raise one lb. of water from 32° to 240° is 209 units, and to raise it from 32° to 382 degrees, the temperature of steam at 200 lbs. pressure requires 355 thermal units. The increase in the sensible heat of the water, therefore, is 355 minus 209 = 146 units, or about the same as the temperature increase for these two pressures, which is 142°. It is thus clear that the total increase in the number of heat units in steam raised from 25 lbs. to 200 lbs. pressure is small (43° as found

above) because the latent heat absorbed in the formation of the steam decreases as the pressure increases. It requires less heat to generate steam from water raised to 382° at 200 lbs. pressure, than from water previously raised to 240° at 25 lbs. pressure. To generate higher pressure steam, therefore, we must first apply enough heat to bring the water to a temperature corresponding to the higher pressure. This heat will be nearly proportionate to the increase in temperature. Then enough heat must be applied to the water to generate the steam, the amount of heat required for this purpose decreasing as the pressure increases. The combined result of these two processes is that it takes only a very small increase in the total heat to produce the higher pressure steam. The idea may be suggested that if this higher pressure is obtained at the cost of so small an expenditure of heat, it would not be reasonable to expect a large gain in economy from it, since it is not possible for the steam to do a greater amount of work than the equivalent of the heat which it contains. This would be true were it not for the fact that the larger part of the heat in the steam is rejected during the exhaust. To illustrate, suppose an engine to exhaust at atmospheric pressure, or at about 15 lbs., absolute, and that the steam is saturated. As may be determined from the steam tables, there would be ejected 1,147 heat units per pound of steam, or 51 heat units less than were found to be in a pound of steam at 200 lbs. pressure. That is to say, under the above assumption, there are available only 51 heat units per pound of steam to do the work in the engine cylinder when the steam pressure is 200 lbs. But we also found that the increase in the heat units in raising the steam pressure from 25 to 200 lbs. was 43, and hence the increase in proportion to the number available is large, although the increase in proportion to the total number required

to generate the steam is small. This shows why high-pressure steam is economical to generate and profitable to use. It should be stated that the only way in which the full benefit can be derived from high pressure-steam is by using the steam expansively, keeping the terminal pressure at release as low as possible. We will not take the space to give the calculations to prove this, but will compare a few results of calculations. Suppose steam to be used in a theoretically perfect engine at the pressure of 25 lbs., 50 lbs., 100 lbs. and 200 lbs. We will assume that in each case the cut-off is at one-third stroke, giving three expansions and a terminal pressure of one-third the initial pressure. The steam consumptions will then be, respectively, about 16½, 16, 15½, and 14 lbs. per horse-power, showing that gain from the increase in pressure is very slight. On the other hand, suppose the expansions to be carried to the atmospheric pressure in each case. The consumptions will then be about 27, 15, 11 and 8 lbs. respectively, showing a marked decrease.

Still another point should be mentioned in relation to the relative gain that is to be expected with the increase in pressure. Comparing the last figure, it will be observed that the decrease in consumption when the pressure increased from 25 to 50 lbs. was 27 minus 15 = 12 lbs., or 44 per cent. Again, when the pressure doubled from 50 to 100 lbs., the consumption decreased only 4 lbs., or 27 per cent; and when the pressure was again doubled to 200 lbs., the consumption only decreased 3 lbs., or about 27 per cent. It is evident from this that the saving from an increase in steam pressure grows less as the pressure increases, and this is found to be the case in actual practice. There is another reason for this, also, coming from the losses incident to cylinder condensation and re-evaporation, which is more marked where there is a wide range in pressures than where the pressures are more uniform throughout the stroke. It is found that where the steam pressure is much above 100 lbs. gauge pres-

sure, no gain will result from a further increase in pressure without compounding, the advantage of the compound engine being that the extremes of temperature in the cylinders are not so great as with a simple engine.

USING STEAM FULL STROKE.

The steam engine is nothing in the world but an enlargement upon the end of the steam pipe, containing a piston against which the steam in the boiler may press. The piston moves a certain distance, and then the steam is allowed to press upon its other side, while the steam on the first side is allowed to flow into the atmosphere and go to waste. The slide-valve is the device ordinarily employed to admit the steam, alternately, to opposite sides of the piston, and to permit the free outflow of steam from the reverse side of the piston. As the steam presses upon the piston, the piston moves forward with a force equal to the pressure of steam per square inch, multiplied by the number of square inches of piston surface. Steam occupies the entire space from the surface of the water in the boiler, to the piston of the engine. The steam space, therefore, includes the steam space of the boiler, the steam pipe, the steam chest, and the cylinder space upon one side of the piston. As the piston moves, the entire steam space becomes a little larger, by reason of the cylinder space becoming longer. Thus it will be seen that all of the steam in the boiler and pipe and engine, would expand a trifle and the pressure become somewhat reduced, were it not for the fact that new steam is made by the fire as fast as the piston moves forward. By this means the steam is maintained at about uniform pressure. It will be seen that the pressure is produced upon the piston by the generation of new steam from the water, that is, the fire causes the water to generate a quantity of steam, and this quantity of steam forces its way into the other steam, exerting a force upon the whole body of steam and pushing the piston ahead.

If an engine piston has a surface of 100 square inches and a stroke of ten inches, it follows that the piston will yield a thousand cubic inches additional steam space by its movement during one stroke, and consequently, the fire will be called upon to produce 1,000 cubic inches of new steam for each single stroke of the engine. If the pressure of the steam be eighty pounds to the square inch, the engine piston will move with the force of 8,000 pounds. When the engine has completed one stroke, we find an amount of power exerted equal to 8,000 pounds moved ten inches, and we then open the exhaust valve and empty into the atmosphere 1,000 cubic inches of eighty-pound steam. We keep on doing this for each stroke. Now our attention is particularly called to the fact that when we empty the steam out of the cylinder, it is just as good as when it went into the cylinder; that is, it was 1,000 cubic inches of steam at a pressure of eighty pounds to the square inch, and when it goes into the atmosphere it will expand into over 6,000 cubic inches, at fifteen pounds pressure to the square inch, or the same pressure as the atmosphere. This 1,000 cubic inches of steam which we dumped out of the cylinder, is precisely the same quality of steam as the steam which we have penned up in the boiler; and which we have to be making new all the time in order to keep the engine running. Such is the operation of the steam engine which receives its steam the full length of the stroke; and such an engine may be described briefly, as a very wasteful machine, which throws away steam as good as it receives it, and which requires the generation of a cylinder full of full pressure steam for each stroke. It should be readily understood that when the piston has completed its stroke, and just before the exhaust valve is opened to allow the steam to escape, the cylinder contains 1,000 cubic inches of steam at eighty pounds pressure, which it is capable of expanding into many thousand cubic inches at constantly decreasing pressure. The first step in the improvement of such an

engine would be to so arrange things as to get some benefit from this enormous power of expansion. The full stroke engine does not get one-half of the power before it throws the steam away. The engine which we would have referred to would yield a power of 8,000 pounds moved ten inches at each single stroke; 33,000 pounds moved one foot in one minute is a horse-power; 66,000 pounds moved half a foot would be the same. An engine using steam full stroke is such an extravagant contrivance that we, now-adays, seldom find them in use. There are certain classes of engines built, fitted with link motions for driving the valve, and they are arranged so as to carry their steam full stroke, but provision is also made for quickly hooking up the link and suppressing the full-stroke feature.

SLIDE VALVE ENGINES.

If we have an engine arranged to receive its steam full stroke and to dump the steam out into the air in as good condition as it was received, and we wish to get some of the benefits of the expansive power of the steam, there is a simple way of doing it and without any great change in the engine, and that is, to lengthen out the slide valve so that after the cylinder is half full of steam, the valve will shut and let no more steam enter. During the balance of the stroke, the entire power comes from the gradual expansion of the steam shut up in the cylinder, and it will be readily seen that whatever power we succeed in getting out of the expansion of the steam, is pure gain. The lower the pressure of the steam is when it is exhausted into the air, the more it has expanded, the more power we have gotten out of it, and the more we have gained. It may be said in a few words, that all slide-valve engines are now arranged to work their steam expansively. But it is, unfortunately, found that the slide-valve possesses a peculiar defect, which prevents the system being carried very far. We can lengthen out a slide-valve so as to cut the

steam off at any desired point of the stroke, and we must then increase the throw of the eccentric in order to properly operate the long valve. But the minute we do this we find that we have interfered, to a certain extent, with the proper operation of the exhaust. No matter what we do about the admission of steam or about cutting off before the end of the stroke, we must arrange our exhaust to take place at a certain point at the end of the stroke. It is found in practical operations that this necessary quality of the slide-valve prevents our arranging it to cut off the steam properly at an earlier point than about five-eighths or three-quarter stroke. The consequence is, that an engine with two feet stroke will receive steam 18 inches, then have 6 in. of expansion. It may be fairly said, in a general way, that about all the slide-valve engines now manufactured, cut off the steam at about five-eighths or three-quarters stroke; and it may be further said that this is about all we can get out of a slide-valve engine. Even the trifling expansion got from such engines as this, represents an immense amount of money in the course of a year in large establishments, but it is not good enough for anyone who seeks even a decent investment of money, in power-getting appliances.

REGULAR EXPANSION ENGINES.

A liberal expansion of steam being desirable and the slide-valve proving totally incapable of providing for such expansion, the first step in the desired direction is to totally discard the slide-valve. The Corliss valve is a cylindrical piece, oscillating in a cylindrical hole. The valve does not fill this hole, but seats against one side only. Hence, the fitting qualities are about the same as with the slide-valve and, in fact, the principle is about the same, the Corliss representing a portion of the slide-valve, rolled into the form of a cylinder and operating in a concave seat. We must not only discard the slide-valve arrangement, but in the valve arrangement which we select, we must secure an abso-

lute independence between the steam admission part of the system and the exhaust part. The slide-valve is one chunk of cast iron, letting in and cutting off steam at its outside edges, and opening and closing the exhaust by its inside edges. 'When one of these valve edges moves, everything else has to move. There is, consequently, no independence of action. In the Corliss engine there are parts to let steam into the cylinder and to quit letting it in at the proper time, and there are valves to let it out at the proper time, and they are perfectly independent of each other in all of their movements. The consequence of this arrangement is, that the steam valve may open, steam flow into the cylinder, the valve suddenly shut and chop the steam off short, the piston move forward in its stroke by the expansion of the confined steam, and finally, be let out by the opening of the exhaust valve, which has all the time stood ready for the discharge. Here we have a regular expansion engine. We can cut the steam off as early in the stroke as we desire, and hence, have any degree of expansion we desire. And we can do this without interfering with the exhaust valves. It is found, in practice, that an engine cutting off at about one-fifth of its stroke and expanding the other four-fifths, will yield the fairest practical economy.

AUTOMATIC CUT-OFF ENGINES.

In order that those not posted may understand what is meant by the term "Automatic Cut-off Engines," we will have to go back a step. Take, for instance, a full-stroke engine. It ought to be well understood how the ordinary governor does its work. Suppose, for instance, that there is no governor, and that we regulate the speed of the engine by having a man stand at the throttle-valve all the time. If the engine runs too fast, he shuts the throttle-valve a little. This makes the steam pipe so small that the steam cannot flow fast enough to keep the pressure up, and consequently,

the speed goes down. If the engine runs too slow, he opens the throttle-valve and lets the steam flow free, so as to maintain higher pressure. Thus it will be seen that the man at the throttle regulates the engine by altering the pressure with which the steam acts upon the engine. An ordinary engine governor is simply a man at the throttle. When the engine runs too fast the balls fly out, the governor valve shuts a little and the pressure of steam entering the engine is reduced, and so on through all the changes continually taking place. All steam engines, in which the regulation of steam is effected by means of a governor operating upon a throttle, are called throttling engines. They operate by reducing the pressure of the steam admitted to the engine, and thereby taking so much of the vitality out of the steam. It is entirely the wrong way to do it. After once spending our money to get up pressure in the boiler, we should make the greatest possible use of that pressure, so long as we are taking the steam from the boiler. It is, therefore, desirable that the full boiler pressure should be admitted to our cylinder; and the question arises as to how we shall be able to regulate the speed if we do not tinker with this pressure. The automatic engine regulates the speed by the simple act of altering the point of cut-off. If the engine is cutting off at one-fifth stroke, we get a power equal to the incoming force of steam for one-fifth of the stroke, and the expansion of the steam for the other four-fifths of the stroke. If the engine runs too slow we cut the steam off a little later and thereby increase the average pressure during the expansion. The automatic engine, then, is an engine, which cuts off the steam at an earlier point in the stroke, if the engine runs too fast, and cuts it off at a later point if it runs too slow. It is the duty of the governor to say just when the steam valve should close and not let any more steam into the cylinder. In the Corliss engine the steam valves open wide at the beginning of the stroke and let full boiler pressure smack in against the piston.

After the piston has advanced to, say one-fifth of its stroke, the valve shuts up as quick as a flash and the expansion begins. If the engine starts too slow, the governor will hold the steam valve open a trifle longer, but will not interfere with its full opening at the beginning of the stroke, or with its flash-like closing when the cut-off is to take place. During all these operations of the governor and the admission valves, the exhaust valves are let entirely alone, and they continue their work unchanged. It will thus be seen that the expansion engine makes provision for the utmost economy in the use of steam, and with the automatic feature added to it, provides that this economy shall not be sacrificed for the purpose of regulating the speed.

THE GARDNER SPRING GOVERNORS.

Construction. — Two balls are rigidly connected to the upper ends of two flat, tapering, steel springs — the lower ends of the springs being secured to a revolving sleeve,which receives rotation through mitre gears ; links connect the balls to an upper revolving sleeve, which is free to move perpendicularly.

The valve stem passes up through a hollow standard upon which the sleeves revolve, and is furnished with a suitable bearing in the upper sleeve ; the closing movement of the valve is upward, and is obtained in the following manner : The balls at the free ends of the springs furnish the centrifugal force and the springs are the main centripetal agency (gravity is not employed). As the balls fly outward, under the centrifugal influence, they move in a curved horizontal path which may be described as an arc, modified by a radius of changing length — the radius being represented by the length and position of the springs ; the links represent a radius of lesser length, while the sleeve to which the lower ends of the links are pivoted, being free to rise and fall, nullifies the effect of the links in determining the arc in which the balls travel. As the

balls move outward in their peculiar path, the sleeve is drawn upward by the links, and, as the balls move inward, the sleeve is pushed downward. The change of speed is obtained by increas-

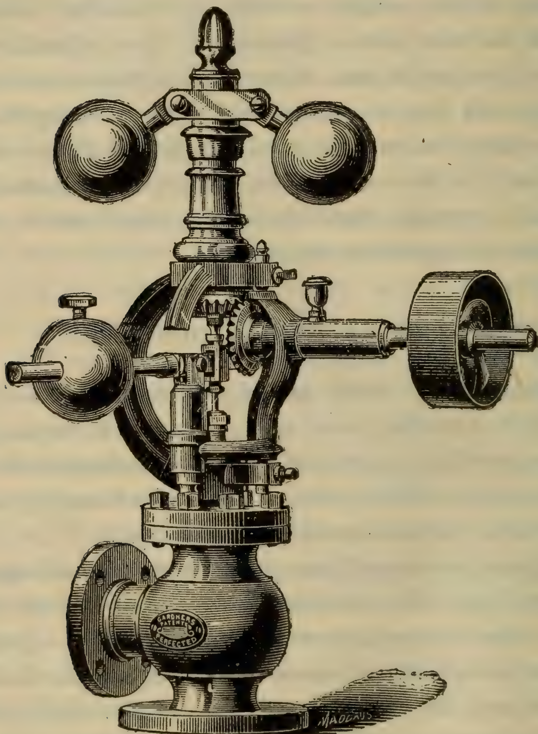


Fig. 198. The Gardner standard governor — class "A" with automatic safety stop and speeder.

ing or decreasing the centripetal resistance, and accomplished by the action of a spiral spring pivoted against the lever, and by means of a shaft and arm against the valve-stem in the direction to open the valve; a thumb-screw is used to adjust the compres-

sion. A convenient sawyer's lever is attached to the shaft, and a reliable automatic safety stop is furnished when desired.

Fig. 198 on the preceding page represents the Gardner Standard Governor, Class "A."

This is a gravity governor, having an automatic safety stop and speeder. It is made in sizes from $1\frac{1}{2}$ inches to 16 in., and is especially adapted to the larger type of stationary engines. In action, the centrifugal force of the pendulous balls is opposed by the resistance of a weighted lever, the speed being varied by the position of the weight. The automatic safety stop is very simple in construction and reliable in action. It is accomplished by allowing a slight oscillation of the shaft bearing, which is supported between centers and held in position by the pull of the belt; a projection at the lower part of the shaft bearing supports the fulcrum of the speed lever. If the belt breaks or slips off the pulley, the support of the fulcrum is forced back, so as to allow the fulcrum to drop and instantly close the valve. The valve is not affected by steam current and both valve and seats are made of special composition, that effectually resists wear and the cutting action of the steam. The governor is made for all pressures, all parts being made by the duplicate system, with special machinery.

Fig. 199 on the following page represents Class "B" governor — a combination of the gravity and spring designs.

They are made in sizes from $\frac{3}{4}$ to 10 inches inclusive, and are adapted to all styles of engines. They are provided with speeder and sawyer's lever, but are not automatic. In the Class "B" governor the centrifugal force of the pendulous balls operates against the resistance of a coiled steel spring, inclosed within a case and pivoted on the speed lever by means of a screw; the amount of compression of the spring can be changed so as to give a wide range of speed. A continuation of the speed lever makes a convenient sawyer's hand lever, which controls the valve by

means of a cord. Sizes $\frac{3}{4}$ to $1\frac{1}{4}$ in., inclusive, have an adjustable frame, which can be set at any desired angle in relation to the

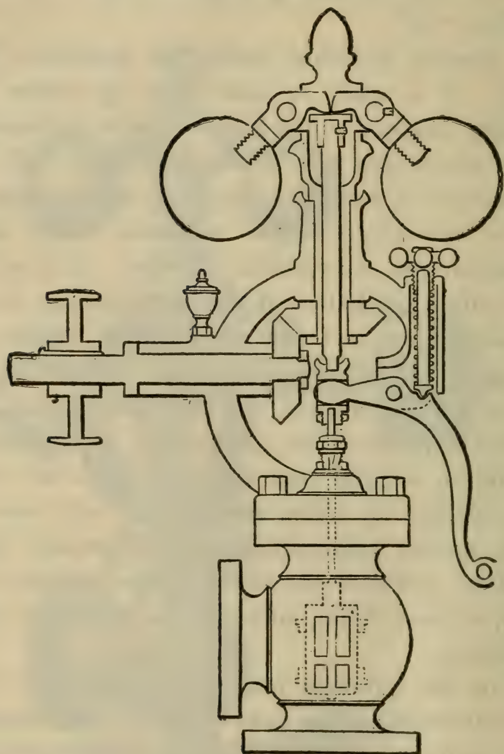


Fig. 199. The Gardner standard governor—Class "B."

valve chamber. The valve and chamber are the same as used on Class "A" governor, and they are made with the same care and style of workmanship.

CHAPTER XIV.

A FEW REMARKS ON THE INDICATOR.

The steam-engine indicator is an instrument designed to show the steam pressure in the cylinder at all points in the stroke. It consists primarily, of a piston of known area capable of moving in a cylinder and resisted by a coil spring of known strength. To this piston is attached, by means of suitable piston rod and levers, a pencil capable of tracing a line corresponding to the motion of the indicator piston. This line is traced on a paper slip attached to the drum of the indicator, which drum is connected to some moving part of the engine in such a way as to have a back and forward movement, coincident with the steam piston of the engine.

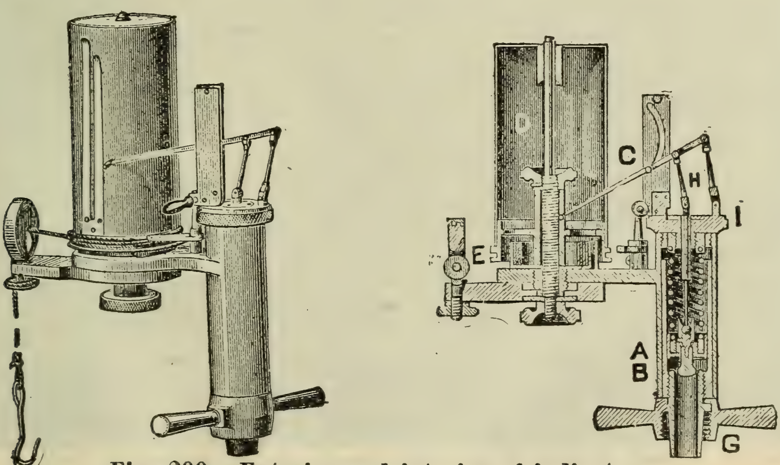


Fig. 200. Exterior and interior of indicator.

By referring to the above sectional view of an indicator, which is generally recognized as the best type, the construction will be readily understood.

THE USE OF THE STEAM ENGINE INDICATOR IN SETTING VALVES AND THE INVESTIGATION OF SOME OF THE DEFECTS BROUGHT OUT BY THE INDICATOR CARDS.

The steam-engine indicator has come into such general use that to-day there are but few men running engines who are not familiar with its construction and manner of attachment to engines, and the method of calculating horse power from cards. The indicator is attached to pipes tapped into the cylinder heads, or into the barrel of the cylinder opposite the counterbore, beyond the travel of the piston rings. The indicator consists of a cylinder with piston and compression spring and a drum attached to a coiled spring, used for returning the same. The pressure of steam on the piston of the indicator compresses the spring above it. The motion of the piston is carried by a piston-rod to a pencil motion, which multiplies the motion of the spring some five or six times. The springs are marked 20, 40, 80, etc. This meaning that 80 lbs. pressure per square inch on the indicator piston (or whatever the spring may be marked) will cause the pencil at the end of the pencil-arm to move an inch. The pencil marks on paper, which is fastened on a drum. This drum is moved by the cross-head of the engine, through some form of reducing motion, such as pantograph, lazy-tongs, brumbo pulley, etc. To obtain the horse power, we first need the mean pressure equivalent to the variable pressure on the card. This is most easily found by dividing the area of the card by the length, giving the height of a rectangular card of equivalent area, and then multiplying this height by the scale of the spring. The mean effective pressure per square inch on the piston, times the area of the piston in square inches, times the speed of the piston in feet per minute, divided by 33,000, gives the horse power. If there is a loop at either end of the card, the area of this loop is to be subtracted from the larger area before finding the mean height of

the card, since such a loop represents work opposed to the working side of the piston. In getting areas by means of a planimeter, no attention need be given to the loops. By following the lines in order, as drawn by the indicator pencil, the loops will be subtracted from the main card, for if the main body of the card is traced in a right-handed rotation, the loops will be traced in a left-handed rotation.

DIAGRAM ANALYSIS.

Figs. 201 and 202 are from throttling engines; the former representing good performances for that class of engine, and the latter,

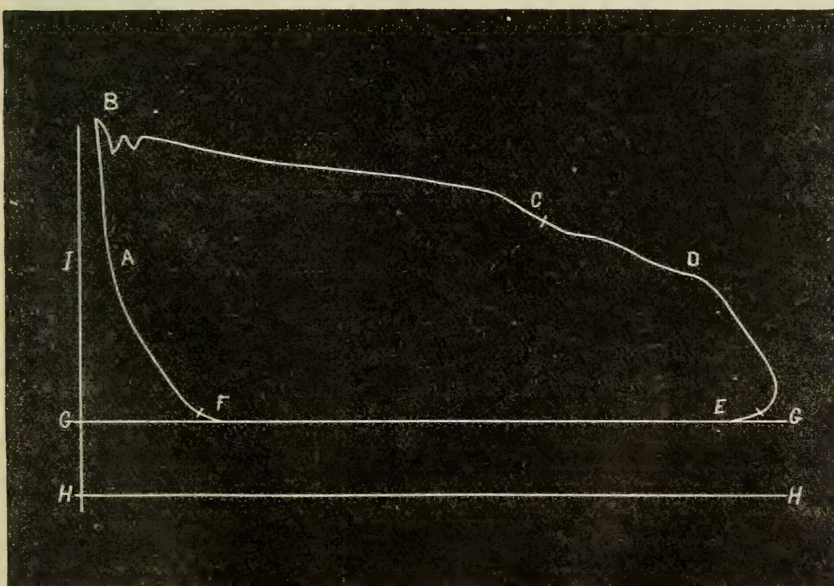


Fig. 201. Diagram from a throttling engine.

in some respects, which the engineer will readily recognize, bad performances.

Figs. 203, 204, and 205, are from automatics; Fig. 203 representing what is now considered rather too light a load for best practical economy; Fig. 204 about the best load, and Fig. 205 is from a condensing engine.

Line *A B* is the induction line, and *B C* the steam line; both together representing the whole time of admission.

C is about the point of cut-off, as nearly as can be determined by inspection. It is mostly anticipated by a partial fall of pressure due to the progressive closure of the valve.

The usual method is, to locate it about where the line changes its direction of curvature.

C D is the expansion curve. *D* is the point of exhaust.

D E is the exhaust line, which begins near the end of the stroke and terminates at the end of the stroke, or, at latest, before the piston has moved any considerable distance on its return stroke.

The principal defect of Fig. 202 is, that this line occupies nearly all the return stroke. *E F* is the back pressure line, which, in non-condensing engines, should be coincident with, or but little above, atmospheric pressure. In Fig. 205 it is below the atmospheric line to the extent of the vacuum obtained in the cylinder. Some authorities would call it the vacuum line in Fig. 205 but that name properly belongs to a line representing a perfect vacuum.

F is the point of exhaust closure (slightly anticipated by rise of pressure) and *F A* the compression curve, which, joining the admission line at *A*, completes the diagram proper, forming a closed figure.

G G is the atmospheric line traced when the piston of the indicator is subject to atmospheric pressure, above and below alike. Some pull the cord by hand when tracing it, to make it longer than the diagram. *H H* is the vacuum line, which, when required, is located by measurement such a distance below the atmospheric line as to represent the atmospheric pressure at the time and place, as nearly as can be ascertained. The mean

atmospheric pressure at the sea level is 14.7 pounds. For higher altitudes, the corresponding mean pressure may be found by multiplying the altitude by .00053, and subtracting the product from 14.7. When a barometer can be consulted, its reading in inches multiplied by .49 will give the pressure in pounds.

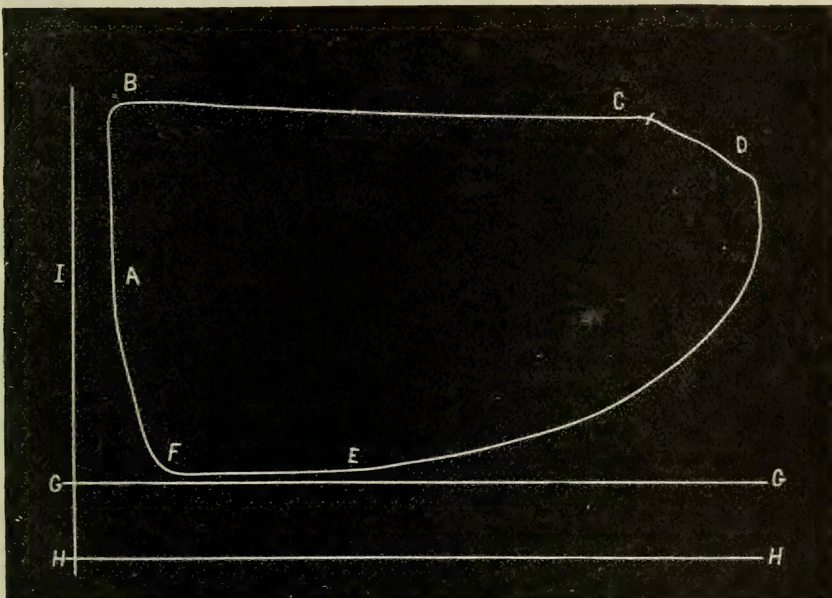


Fig. 202. Diagram from a throttling engine.

I is the clearance line, representing by its distance from the nearest point of the end of the diagram at the admission end, as compared with the whole length, the whole volume of clearance known to be present. Its use is mainly to assist in constructing a theoretical expansion curve by which to test the accuracy of the actual one.

Calculating mean effective pressure. — Since the simplification and popularization of the planimeter, no engineer who has occa-

sion to compute the "indicated horse-power" (IHP) of engines should be without one; for, if properly handled, the results

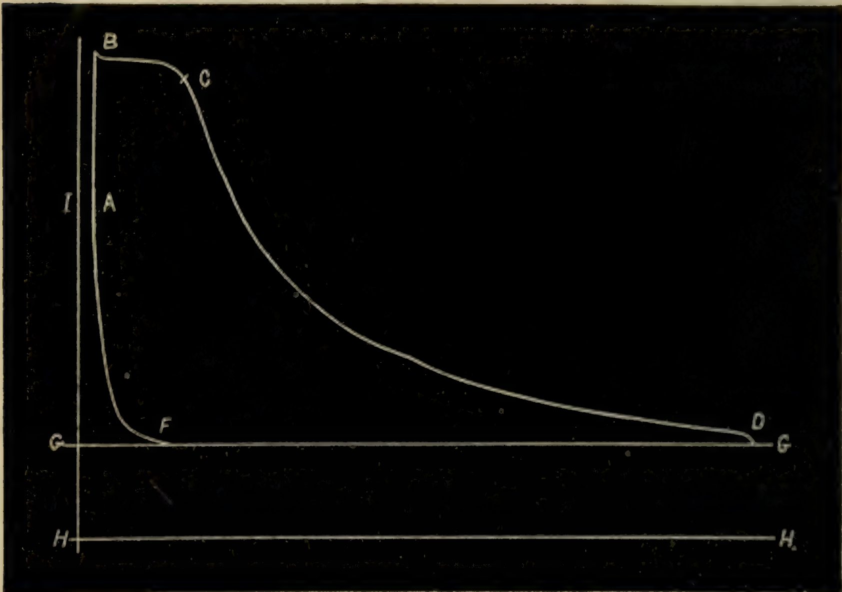


Fig. 203. Diagram from an automatic cut-off engine.

obtained by them are more accurate and more quickly obtained than by any other process. The diagram is pinned to a smooth board covered with a sheet of smooth paper, the pivot of the leg pressed into the board at a point which will allow the tracing point to be moved around the outline of the diagram without forming unnecessarily extreme angles between the two legs, and a slight indentation made in the line at some point convenient for beginning and ending; for it is vitally important that the beginning and ending shall be at exactly the same point. The reading of the wheel is taken, or it is placed at zero, and the tracing point is

passed carefully around the diagram, following the lines as closely as possible, moving right-handed, like the hands of a watch. The reading obtained (by finding the difference between the two, if the wheel has not been placed at zero) is the area of the diagram in square inches, which, multiplied by the scale of the diagram, and divided by its length in inches, gives the mean effective pressure.

The process of finding the mean effective pressure by ordinates. — Divide the diagram into 10 equal parts as shown by the full lines in Fig. 204: when performing this work a frequent mistake is made, viz.,

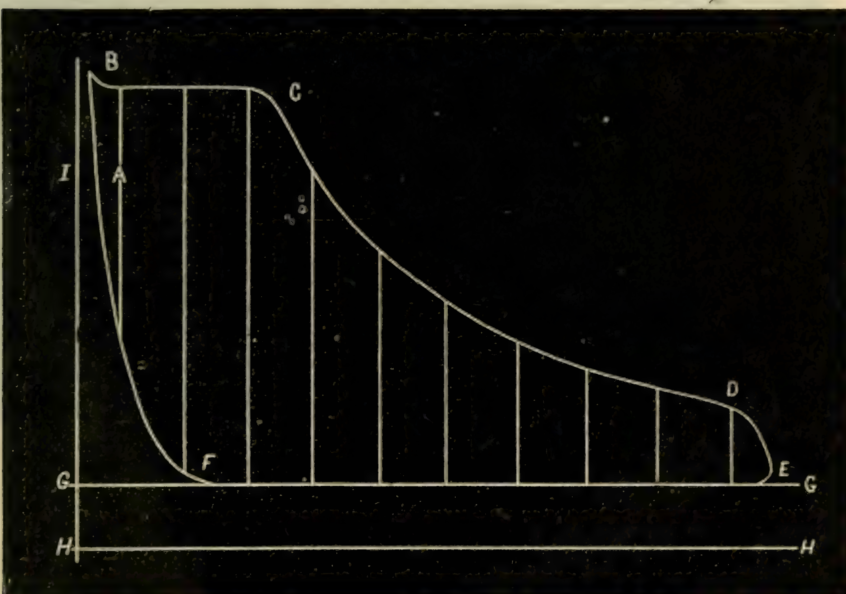


Fig. 204. Erecting the ordinates.

making all the spaces equal. The end ones should be half the width of the others, since the ordinates stand for the centers of

equal spaces. Ten is the most convenient and usual number of ordinates, though more would give more accurate results. The aggregate length of all the ordinates (most conveniently measured consecutively on a strip of paper) divided by their number, and multiplied by the scale of diagram, will give the mean effective

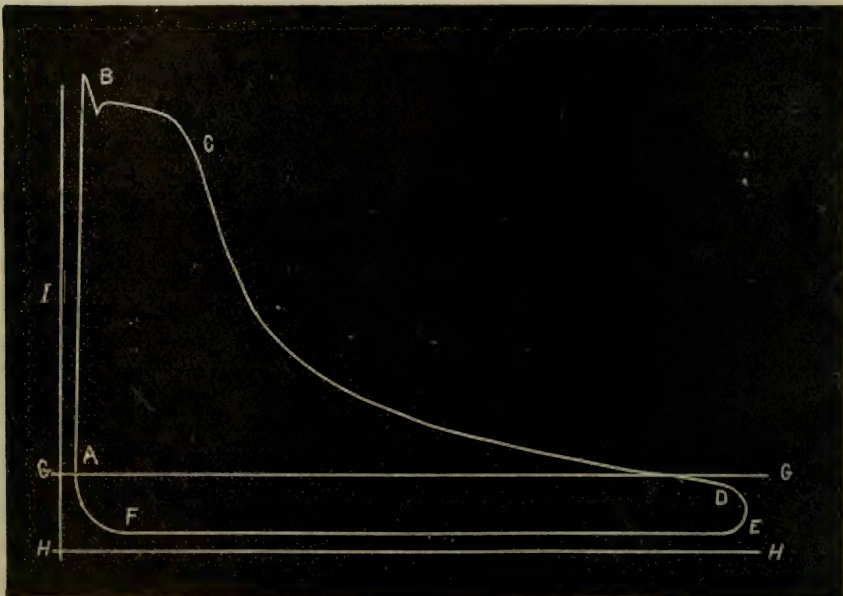


Fig. 205. Diagram from a condensing engine.

pressure. A quick way of making a close approximation to the mean effective pressure of a diagram is, to draw line *a b*, Fig. 206, touching at *a*, and so that space *d* will equal in area spaces *c* and *e*, taken together, as nearly as can be estimated by the eye. Then a measure, *f*, taken at the middle, will be the mean effective pressure. With a little practice, verifying the results with the planimeter, the ability can soon be acquired to make estimates in

this way with only a fraction of a pound of error with diagrams representing some degree of load. With very high initial pressure and early cut-off, it is not so available.

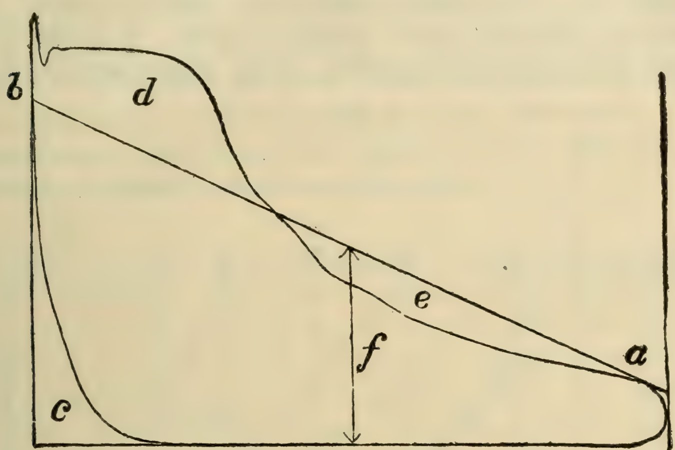


Fig. 206. Method of estimating the mean pressure.

The indicated horse-power. — IHP is found by multiplying together the area of the piston (minus half the area of the piston-rod section, when great accuracy is desired), the mean effective pressure and the travel of the piston in feet per minute, and dividing the product by 33,000. It is sometimes convenient to know the HP constant of an engine, which is the HP for one revolution at one pound mean effective pressure. This multiplied by the mean effective pressure, and by the number of revolutions per minute, gives the IHP.

THEORETICAL CURVE.

Testing expansion curves. — It is customary to assume that steam, in expanding, is governed by what is known as Mariotte's law, according to which its volume and pressure are inversely pro-

portional to each other. Thus, if a cubic foot of steam at, say, 100 pounds pressure be expanded to 2 cubic feet, its pressure will fall to 50 pounds, and proportionately for all other degrees of expansion. The pressures named are "total pressures;" that is, they are reckoned from a perfect vacuum. A theoretic expansion curve which will conform to the above theory may be

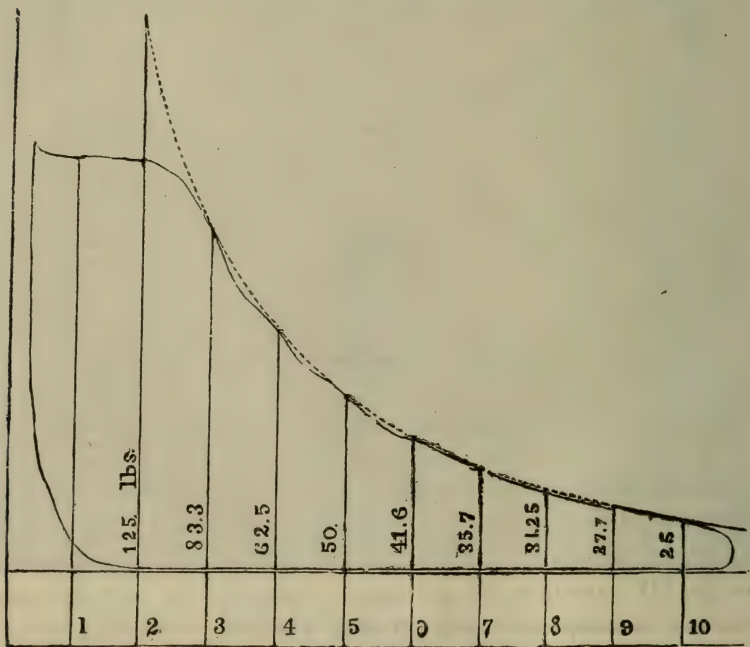


Fig. 207. Locating the true expansion curve.

traced by the following method: Referring to Fig. 207, having drawn the clearance and vacuum lines as before explained, draw any convenient number of vertical lines, 1, 2, 3, 4, 5, etc., at equal distances apart, beginning with the clearance line and number them as shown. Decide at what point in the expansion curve

of the diagram we desire the theoretic curve to coincide with it. Suppose we choose line 10, on which we find the indicated pressure to be 25 pounds. Multiply this pressure by the number of the line (10) and divide the product (250) by the numbers of each of the other lines in succession. The quotients will be the pressures to be set off in the lines. Thus, 250 divided by 9 gives 27.7, the pressure on line 9; and so for all the others. The same curve may also be traced by several geometric methods, one of which is as follows, referring to Fig. 208: —

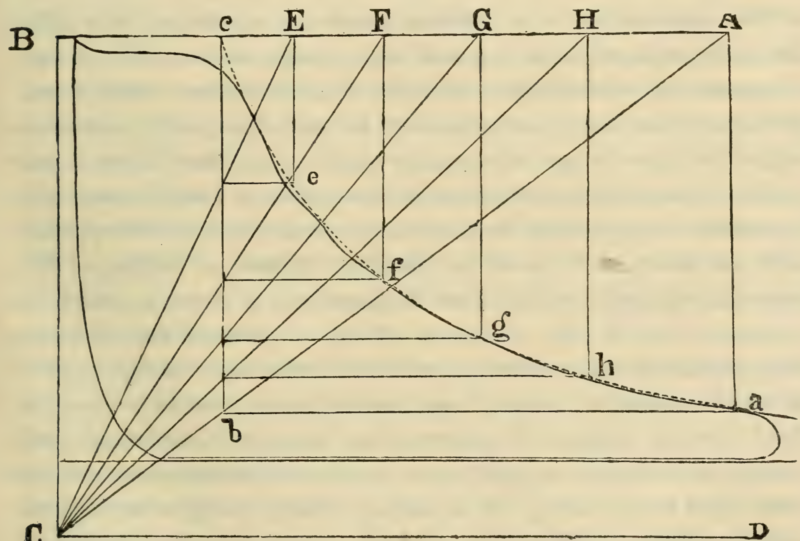


Fig. 208. Drawing the hyperbolic curve.

Having drawn the clearance and vacuum lines as before, select the desired point of coincidence, as a, from which draw the perpendicular a A. Draw A B at any convenient height above or near the top of the diagram, and parallel to the vacuum line D C. From A draw A C and from a draw a b parallel to D C, and from

its intersection with $A\ B$, erect the perpendicular $b\ c$, locating the theoretical point of cut-off on $A\ B$. From any convenient number of points in $A\ B$ (which may be located without measurement) as E, F, G, H, draw lines to C, and also drop perpendiculars $E\ e$, $F\ f$, $G\ g$, $H\ h$, etc. From the intersection of $E\ C$ with $b\ c$, draw a horizontal to e, and the same for each of the other lines $F\ C$, $G\ C$, $H\ C$; establishing points e, f, g, h, in the desired curve. Any desired number of points may be found in the same way. But this curve does not correctly represent the expansion of steam. It would do so if the steam during expansion remained or was maintained at a uniform temperature; hence, it is called the isothermal curve, or curve of same temperature. But, in fact, steam and all other elastic fluids fall in temperature during expansion, and rise during compression; and this change of temperature augments the change of pressure slightly; so that if, as before assumed, a cubic foot of steam at 100 pounds total pressure be expanded to two cubic feet, the temperature will fall from nearly 328° to about 278°, and the pressure instead of falling to fifty pounds, will fall a trifle below 48 pounds. A curve in which the pressure due to the combined effects of volume and resulting temperature is represented, is called the adiabatic curve, or curve of no transmission; since, if no heat is transmitted to or from the fluid during change of volume, its sensible temperature will change according to a fixed ratio, which will be the same for the same fluid in all cases. It is not necessary to give any of the usual methods of tracing the adiabatic curve, since the isothermal curve is the one generally used for that purpose. And while it is incorrect in that it does not show enough change of pressure for a given change of volume, the great majority of actual diagrams are still more incorrect in the same direction; so that when a diagram conforms to it as closely as the one used in these illustrations, it is considered a remarkably good one. A sufficiently close approximation to the adiabatic curve to enable the non-profes-

sional engineer to form an idea of the difference between the two, may be produced by the following process: Taking a similar diagram to those used for the foregoing illustrations, we fix on a point A near the terminal, where the total pressure is 25 pounds. As before, this point is chosen in order that the two curves may coincide at that point. Any other point might have been chosen for the point of coincidence; but a point in that vicinity is generally chosen so that the result will show the amount of power that should be obtained from the existing terminal. This point is 3.3 inches from the clearance line, and the volume of 25 pounds is 996; that is, steam at that pressure has 996 times the bulk of water. Now, if we divide the distance of A from the clearance line by 996, and multiply the quotient by each of the volumes of the other pressures indicated by similar lines, the products will be the respective lengths of the lines measured from the clearance line, the desired curve passing through their other ends. Thus, the quotient of the first, or 25-pound pressure line divided by 996 is .003313; this multiplied by 726, the volume of 25-pound pressure, gives 2.4, the length of the 25-pound pressure line; and so on for all the rest.

Fig. 209 shows a card taken from a Corliss engine, running at a speed of about ninety revolutions per minute. On account of the slow speed and the quick admission obtained by this form of valve gear, but little compression is needed. For high speed engines, there is much more compression. At high speeds, the expansion line of the indicator card, instead of being a smooth curve like that shown in Fig. 209, is often a wavy line, due to oscillations of the spring in the indicator.

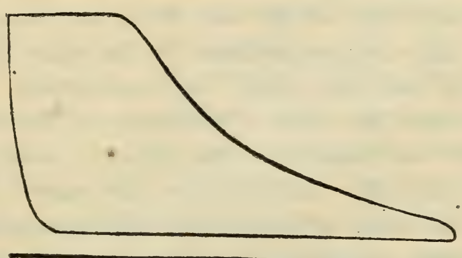


Fig. 209. Diagram from Corliss engine.

Fig. 210 represents what is called a stroke card. The indicator shows us the pressure on one side of the piston for a revolution. When we calculate the horse-power from a card, we are assuming that the back pressure and compression line on the other side of the piston are the same as shown on the card. This may or may not be the case. In calculating the total horse-power for the two ends of the cylinder, any error from this cause affecting the calculation for one end of the cylinder, will be nearly balanced by an opposite error in the calculations for the other end, so that the final result is practically correct. If it were not for the piston-rod making the area of one side of the piston smaller than on the other, there would be absolutely no error arising from this. The stroke card shows the pressure on opposite sides of the piston at all points of the stroke. The difference between the lines at any point is the effective push per square inch. This card is constructed by using the steam and expansion lines of the card from one end, and the back pressure and compression lines for the same stroke, from the card taken on the other end. In constructing diagrams for very accurate work, the ratio of the areas of the two sides of the piston have to be considered; the pressure above the atmosphere for one side being multiplied by this ratio. It will be seen that up to the point of cut-off, the difference of pressure, or effective pressure, is nearly constant; this difference grows less, due to the drop along the expansion curve, till at the point where the two lines cross, the pressure on the two sides balances. Beyond this point, the pressure exerted to hold the piston back

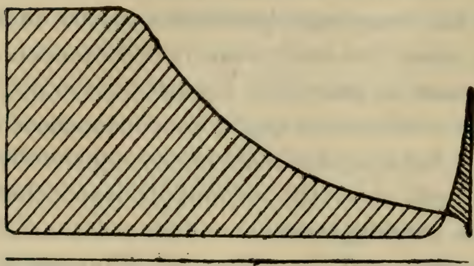


Fig. 210. Showing a stroke card.

is greater than that exerted to push it ahead. The energy stored in the fly-wheel during the first part of the stroke is given out here near the end of the stroke to help the engine over the dead point.

STEAM CHEST CARDS.

By attaching one indicator to the steam chest of an engine, and another to one end of the cylinder, it can be seen whether the pipes and ports are of sufficient size. A sloping steam line on an indicator card may be due to too small a steam pipe, or too small steam ports, or to both of these combined. This does not apply, of course, to engines using throttling governors.

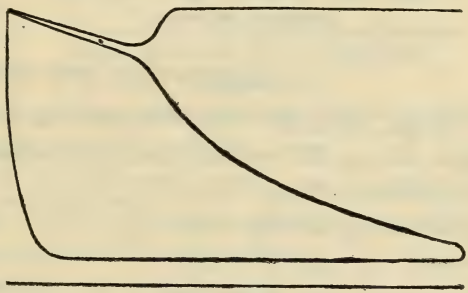


Fig. 211. Steam chest card on forward stroke.

Fig. 211 shows the effect of too small steam pipe. When steam is admitted to the cylinder, there is a drop in pressure in the chest. This drop becomes greater in amount as the speed of the piston increases. At cut-off, the flow of steam into the cylinder stops, then the pressure in the chest reaches boiler pressure. If there is no great drop in the line on the steam chest card, and a considerable drop in the steam line of the card, it would mean that the ports are too small. Such a case is shown by Fig. 212.

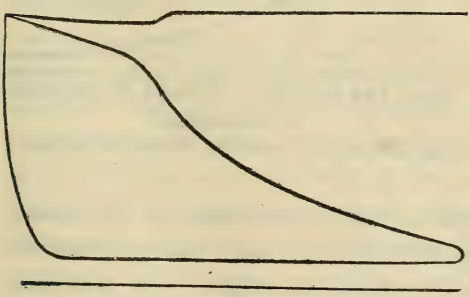


Fig. 212. Steam chest card on forward stroke.

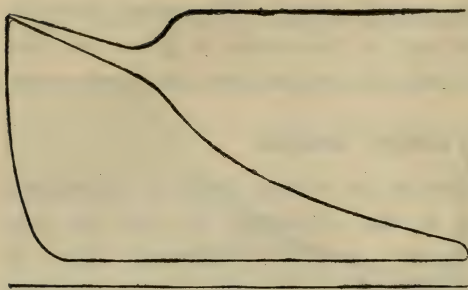
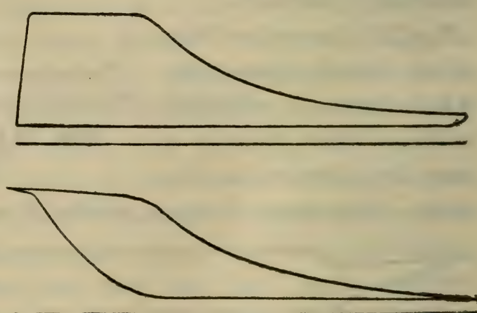


Fig. 213. Steam chest card on forward stroke.

If there is a drop in the chest line up to cut-off, and a still greater drop in the steam line of the card, it would indicate that both the steam ports and the steam pipe were too small. Fig. 213 shows such a case.

ECCENTRIC OUT OF PLACE.

Figs. 214 to 217 inclusive, show cards taken from a Corliss engine having the eccentric out of adjustment. Similar cards would be obtained from any engine having all the valves moved by one eccentric. The plain slide valve and the locomotive, especially in full gear, would give similar cards for the same derangements of eccentric.

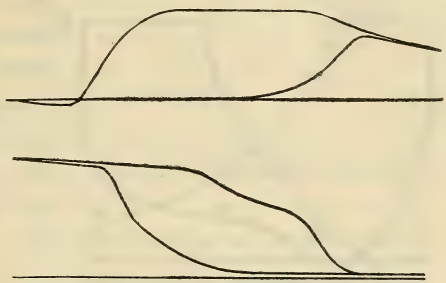


Figs. 214 and 215. Effects of position of eccentric.

Fig. 214 was taken with the eccentric a trifle less than 90° ahead of the crank, or about 20° behind where it belongs on this particular engine.

Fig. 215 shows the eccentric moved too far ahead of the crank. By comparison with Fig. 209, it will be seen that moving the eccentric back makes all the events of the stroke, such as admission, release and compression and cut-off, in the case of engines without automatic cut-off governor, come later; while moving the eccentric ahead brings these events earlier.

Figs. 216 and 217 are similar to Figs. 214 and 215, the only difference being that eccentric is moved a greater distance out of place.



Figs. 216 and 217. Eccentric diagrams.

In Fig. 216 the admission is very late. Release does not occur until after the piston has started on the return stroke, the steam, until released, being compressed back along the expansion curve. This compression is always a trifle below the expansion line, due to the fact that some of the steam has condensed in the interval between the end of the stroke and the release.

Fig. 217 shows too much compression and too early a release. Steam is compressed above boiler pressure in the cylinder, when the valve lifts and the steam escapes into the chest.

Cards like Figs. 214 and 215 are very common.

ECCENTRIC DIAGRAMS.

As small distances near the ends of the indicator cards represent a large angular motion of the crank, the events occurring at the ends of the card are so squeezed together that it is hard to tell from the card just what any peculiarity in the lines may be due to. The eccentric rod working the valves of the engine will be moving at its greatest speed when the crank is near the centers and the piston near the ends of the stroke; since the eccentric is about 90° ahead of the crank. If the motion of the indicator drum is taken from the eccentric rod instead of the cross-head, the card will be changed in shape, compression and release coming near the middle of the card; these are spread out over considerable length, the cut-off, expansion and back pressure lines coming near the ends of the card.

Fig. 218 gives a steam card drawn, assuming that the expansion and compression lines are hyperbolic. The eccentric card for this has been plotted, and corresponding points marked with the same letters. The compression curve, extending from F to A, is a double curve. Admission occurs at A, cut-off at B, release at C, and compression at F.

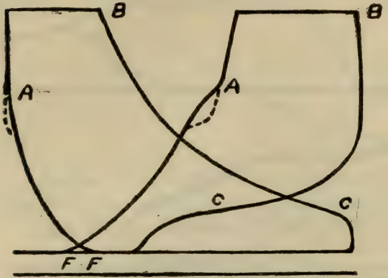


Fig. 218. Combined diagram.

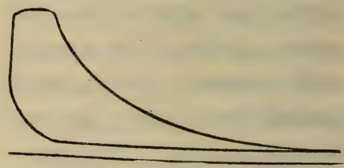


Fig. 219. Ordinary diagram.

Figs. 219 and 220 show cards taken from an engine having tight valves and a tight piston. Corresponding points on the two cards are lettered the same. For a cut-off later than half stroke, the steam line on the eccentric card doubles on itself, as shown in Figs. 220 and 222.

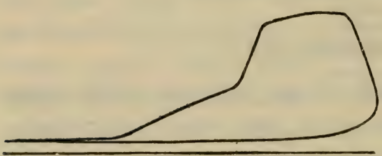


Fig. 220. Eccentric diagram.

The peculiar bend shown by the dotted lines on compression curve of the steam card, Fig. 218, is developed on the eccentric card into a well marked flat place. Evidently this represents a loss of pressure at this point, which may be attributed to one or more of three causes: first, leakage by the piston; second, leakage by the exhaust valves; third, a rapid condensa-

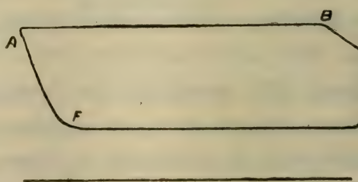


Fig. 221. Eccentric diagram.

tion of steam. If a leakage, it is probable that there is steam blowing by all through the stroke. Near the end of the stroke the piston is moving at so slow a rate that the leakage overbalances the compression. It frequently happens that the pressure drops off at the end of compression, making the upper end of the compression line resemble an inverted letter *U*. If the leakage is by the piston, it will appear or may be made to appear near release, as will be explained later. The effect of compressing steam is to dry it, or, if dry already, to superheat it. While it may be possible in some cases for some of the drop here to be due to condensation, in the majority of cases leakage is the trouble.

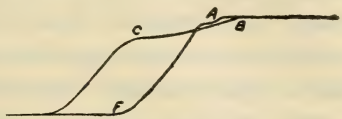


Fig. 222. Eccentric diagram.

Fig. 223 shows the effect of a bad leakage by the piston. This leakage is made evident by the appearance of the upper end of the compression curve and by the increase in pressure along the expansion line just before release. By referring to the stroke card, it will be seen that near this point the pressures on the opposite side of the piston are the greater, so that the leakage is now into the side on which the card is being taken. Unless compression on one side comes earlier than release on the other side, this method would fail. In most engines the valves are set so that compression does come earlier, and all four valve engines can be easily set so as to delay release on one end, and to hasten compression on the other end. In the case of a Corliss engine, this means simply

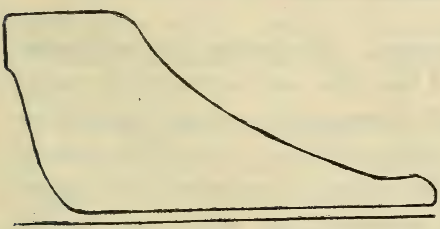


Fig. 223. Effects of leakage.

the changing the length of the rods leading from the wrist-plate to the valve arm. This change can be made with the engine running. It is possible that a card like Fig. 223 might be obtained from a four-valve engine having a leaky steam valve on one end and a leaky exhaust valve on the other end.

Fig. 224 represents the head end and the crank end cards taken from a plain slide valve engine. The valve has equal steam lap and equal exhaust lap. The only trouble in this case

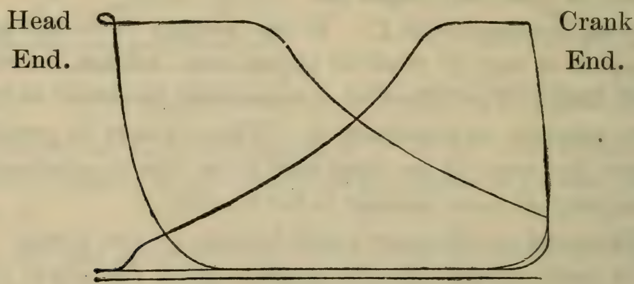


Fig. 224. Effects of changing length of valve stem.

is that the valve spindle is too short. Shortening the valve spindle decreases the outside lap of the valve and increases the inside lap for the head end side, and increases the outside lap and decreases the inside lap for the crank end side. As will be seen by the cards, the head end has the cut-off lengthened, the release delayed, and the compression hastened; the crank-end has the cut-off shortened, the release hastened, and the compression delayed. If the valve spindle were too long the cards shown would be interchanged, the crank end card being the one marked head end.

THE STEAM ENGINE INDICATOR.

Benefits derived and information ascertained from its use. — The benefits derived, and the information ascertained from the use of the steam-engine indicator are varied and important.

The office of the indicator is to furnish a diagram of the action of the steam in the cylinder of an engine during one or more revolutions of the crank, from which is deduced the following data: Initial pressure in cylinder; piston stroke to cut-off; reduction of pressure from commencement of piston stroke to cut-off; piston stroke to release; terminal pressure; gain in economy due expansion; counter pressure, if engine is worked non-condensing; vacuum as realized in the cylinder, if engine is worked condensing; piston stroke to exhaust closure, usually reckoned from zero point of stroke; value of cushion; effect of lead and mean effective pressure on the piston during complete stroke. The indicator diagram, when taken in connection with the mean area and stroke of piston and revolution of crank for a given length of time, enables us to ascertain the power developed by engine; and when taken in connection with the mean area of piston, piston speed and ratio of cylinder clearance, enables us to ascertain the steam accounted for by the indicator.

The mean power developed by engine compared with the steam delivered by boilers, furnishes cost of power in steam, and when compared with the coal, furnishes cost of the power in fuel.

The diagram also enables us to determine with precision the size of steam and exhaust ports necessary, under given conditions, to equalize the valve functions; to measure the loss of pressure between boiler and engine; to measure the loss of vacuum between condenser and cylinder; to determine leaks into and out of the cylinder; to determine relative effects of jacketed and unjacketed cylinders; and to determine effects of expansion in one cylinder, and in two or more cylinders.

TO TAKE A DIAGRAM.

Connecting cord. — The indicator should be connected to the engine cross-head by as short a length of cord as possible. Cord

having very little stretch, such as accompanies the instrument, should be used; and in cases of very long lengths, wire should be used. The short piece of cord connected with the indicator is furnished with a hook; and at the end of the cord, connected with the engine, a running loop can be made by means of the small plate sent with each instrument; by which the cord can be adjusted to the proper length, and lengthened or shortened as required.

Selecting a spring. — It is not advisable to use too light a spring for the pressure. Two inches are sufficient for the height of diagram, and the instrument will be less liable to damage if the proper spring is used. The gauge pressure divided by 2 will give the scale of spring to give a diagram two inches high at that pressure.

To attach a card. — This may be done in a variety of ways, either by passing the ends of it under the spring clips, or by folding one end under the left clip, and bringing the other end around under the right; but, whatever method is applied, care should be taken to have the card rest smoothly and evenly on the paper drum. Now attach the cord from the reducing motion to the engine; but be certain the cord is of the proper length, so as to prevent paper drum from striking the inner stop in drum movement on either end of the stroke.

Tension of drum spring. — The tension of the drum spring should be adjusted according to the speed of the engine; increasing for quick running, and loosening for slower speeds.

The steam should not be allowed into the indicator until it has first been allowed to escape through the relief on side of cock, to see if is clean and dry. If clean and dry, allow it into the indicator, and allow piston to play up and down freely.

Before taking diagram, turn the handle of cock to a horizontal position, so as to shut off steam from piston, and apply pencil to the paper to take the atmospheric line.

In applying pencil to the card, always use the horn-handle screw, to regulate pressure of pencil upon paper to produce as fine a line as possible. After the atmospheric line is taken, turn on steam, and press the pencil against card during one revolution.

When the load is varying, and the average horse-power required, it is better to allow the pencil to remain during a number of revolutions, and to take the mean effective pressure from the average of the several diagrams.

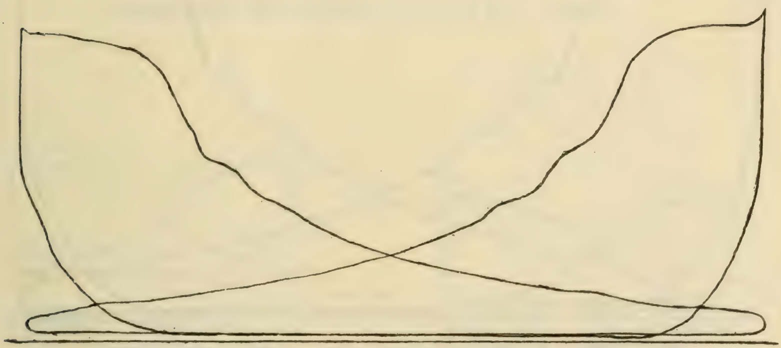


Fig. 225. Diagram from a Russell engine.

Fig. 225 was taken from a Russell engine 13″ x 20″, running 205 revolutions per minute, boiler pressure 98 lbs., scale of spring 60 lbs. Duty, electric lighting.

After sufficient number of diagrams have been taken, remove the piston, spring, etc., from the indicator, while it is still upon the cylinder; allow the steam to blow for a moment through the indicator cylinder; and then turn attention to the piston, spring, and all movable parts, which may be thoroughly wiped, oiled and cleaned. *Particular attention* should be paid to the springs, as their accuracy will be impaired if they are allowed to rust; and great care should be exercised that no grit or substance be introduced to cut the cylinder, or scratch the piston. Be careful

always not to bend the steel bars or rods. The heat of the steam blown through the cylinder of the indicator will be found to have dried it perfectly, and the instrument may be put together with the assurance that it is all ready for use when required. It is a saving of time to keep indicator in order. Any engineer can easily perform the operation without further instruction.

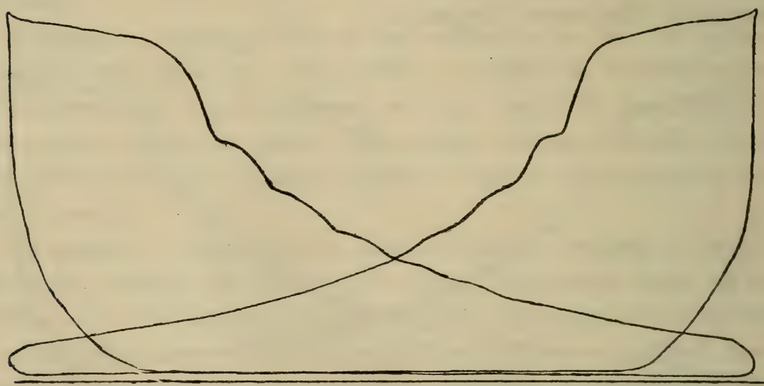


Fig. 226. Another diagram from Russell engine.

Fig. 226 was taken from a Russell engine 16″ x 24″, running 157 revolutions per minute, boiler pressure 70 lbs., scale of spring 40 lbs. Duty, flouring mill.

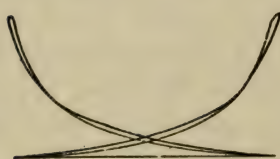


Fig. 227. Friction load.

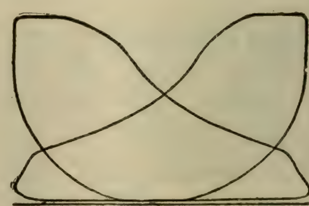


Fig. 228. Full load.

HARRISBURG IDEAL SIMPLE SINGLE VALVE ENGINE.

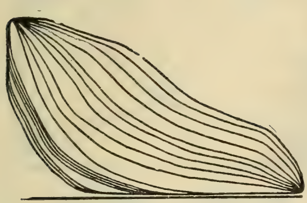


Fig. 229. Graduated load.

Fig. 230. Extreme load variation.

HARRISBURG IDEAL SIMPLE SINGLE VALVE ENGINE.

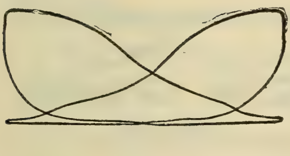


Fig. 231. High pressure diagrams.

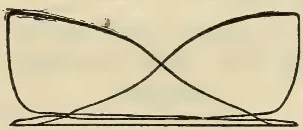


Fig. 232. Low pressure diagrams.

HARRISBURG IDEAL COMPOUND SINGLE VALVE ENGINE.

Fig. 233. Friction diagrams.

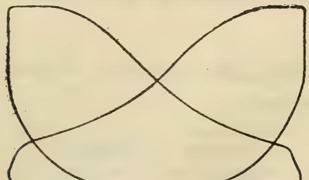


Fig. 234. Full load diagrams.

HARRISBURG STANDARD SIMPLE SINGLE VALVE ENGINE.

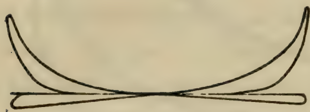


Fig. 235. Friction load.

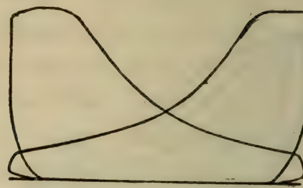


Fig. 236. Full load.

HARRISBURG STANDARD SIMPLE FOUR-VALVE ENGINE.

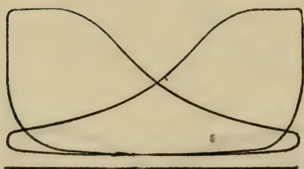


Fig. 237. High pressure diagrams.

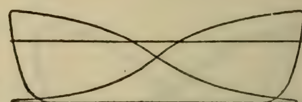


Fig. 238. Low pressure diagrams.

HARRISBURG STANDARD COMPOUND FOUR-VALVE ENGINE.

The indicator diagrams from Fig. 227 to 238 were taken from the Harrisburg Ideal and Standard engines. An engineer will see from these cards the kind of card he should get from a high speed engine of this class.

Fig. 239 is from a Frick Corliss engine, driving a Frick compressor: —

Steam Cylinder	19″ x 28″.
Steam	95 lbs.
Revs.	58
Cond. Press.	164 lbs.
Back Press.	27 lbs.

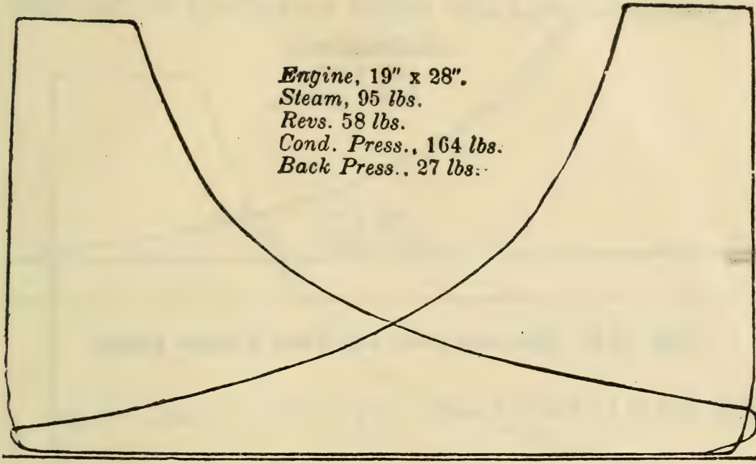


Fig. 239. Diagram from 19" x 28" Eclipse Corliss.

INDICATOR DIAGRAMS FROM 50-TON "ECLIPSE" MACHINE.

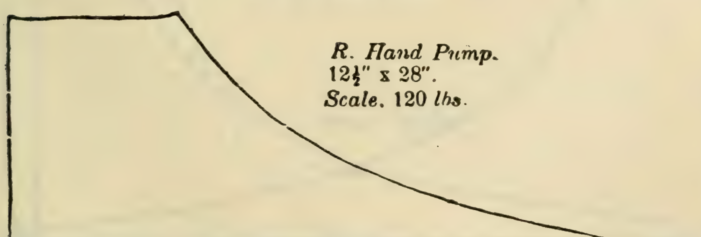


Fig. 240. Diagram from right hand Eclipse pump.

Fig. 240 is R. Hand Pump. 12½" x 28". Scale, 120 lbs.

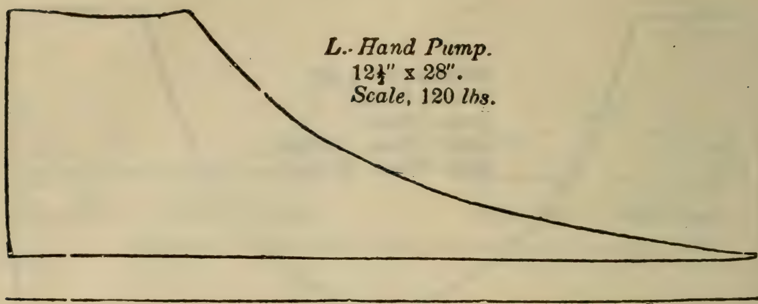


Fig. 241. Diagram from left hand Eclipse pump.

Fig. 241 is L. Hand Pump. 27½″ x 28″. Scale, 120 lbs

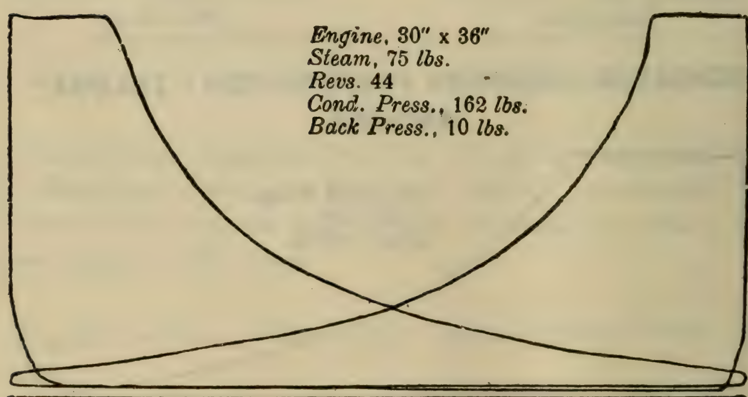


Fig. 242. Diagram from 30″ x 36″ Eclipse machine.

Figs. 242 to 244, are diagrams from a 100-ton "Eclipse" machine.

INDICATOR DIAGRAMS FROM 100-TON "ECLIPSE" MACHINE.

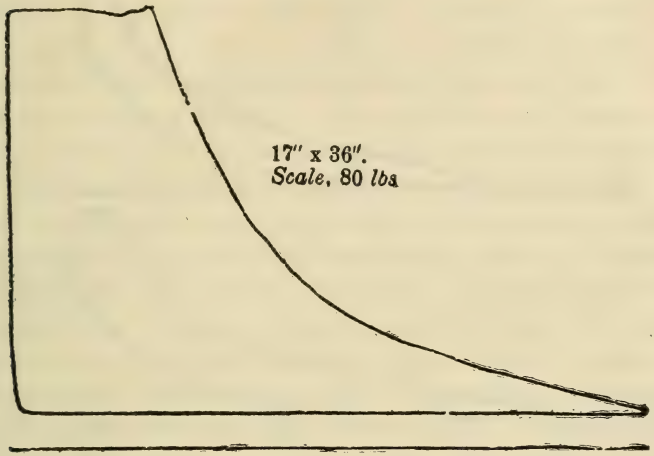


Fig. 243. Diagram from right hand pump.

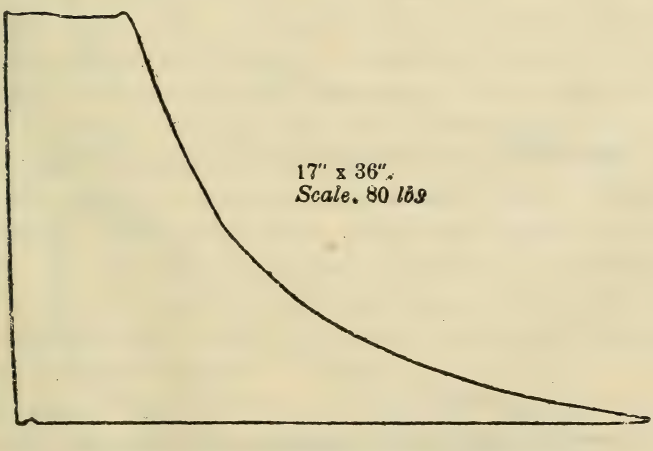


Fig. 244. Diagram from left hand pump.

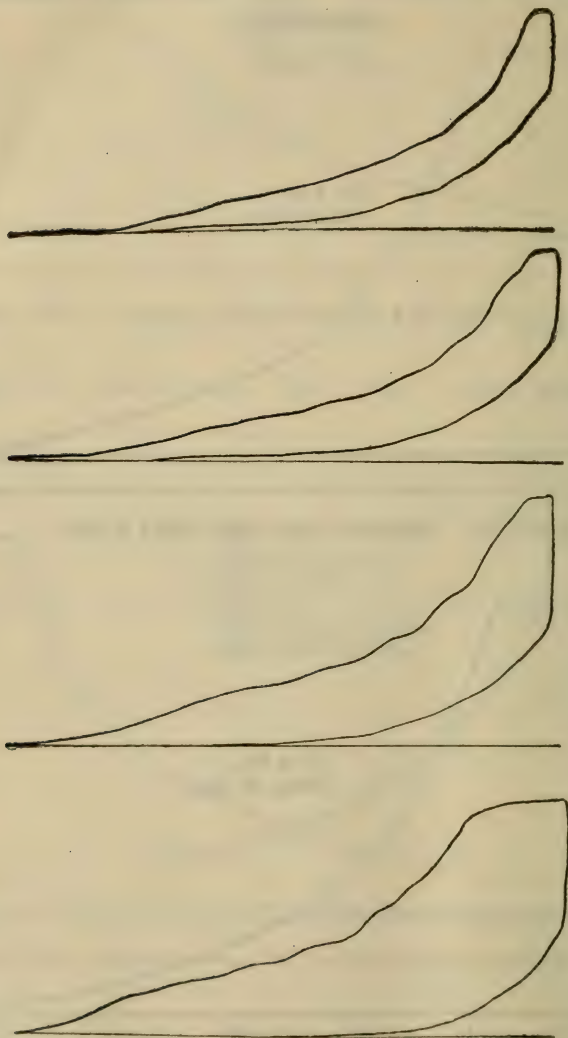


Fig. 245. Diagrams from a Ball engine.

It will be interesting to note that when the eccentric is simply moved forward or backward around the shaft by the action of the governor, all the events of the stroke — admission, release, cut-off and compression — will be hastened or retarded together; but if the eccentric be so designed that the governor will shift it across the shaft instead of around it, the admission and release will be effected differently, and in the opposite direction from the cut-off and compression. If, for example, the cut-off is made to occur earlier in the stroke, the compression will occur earlier also, but the admission and release will occur later instead of earlier. By combining the two movements of the eccentric and having the governor move it partly around and partly across the shaft, it is possible to keep the admission and release nearly constant, while the cut-off and compression vary. This result is attained to a certain extent in the best single-valve engines. Besides these two types, there are numerous other styles of engines in which the point of cut-off is varied automatically. Instead of a shaft governor with a shifting eccentric, a weighted pendulum governor is sometimes employed to operate the link, or radius rod of some one of the various link motions. Sometimes there are separate admission and exhaust valves, the former being under the control of a shaft governor, and the latter operated by a fixed eccentric, so that the points of admission and cut-off only are varied, while the points of release and compression, which depend upon the exhaust valve, remain fixed. There are a great many modifications of the Corliss engine, as originally constructed by Geo. H. Corliss, and there are many engines which, while not resembling the Corliss engine, have some arrangement whereby the cut-off valves are tripped.

On pages 374 and 376 is a collection of diagrams, which illustrate very nicely the peculiarities and difference in the action of throttling and automatic engines. The four diagrams on page 374 were taken from a Ball automatic, in an electric light

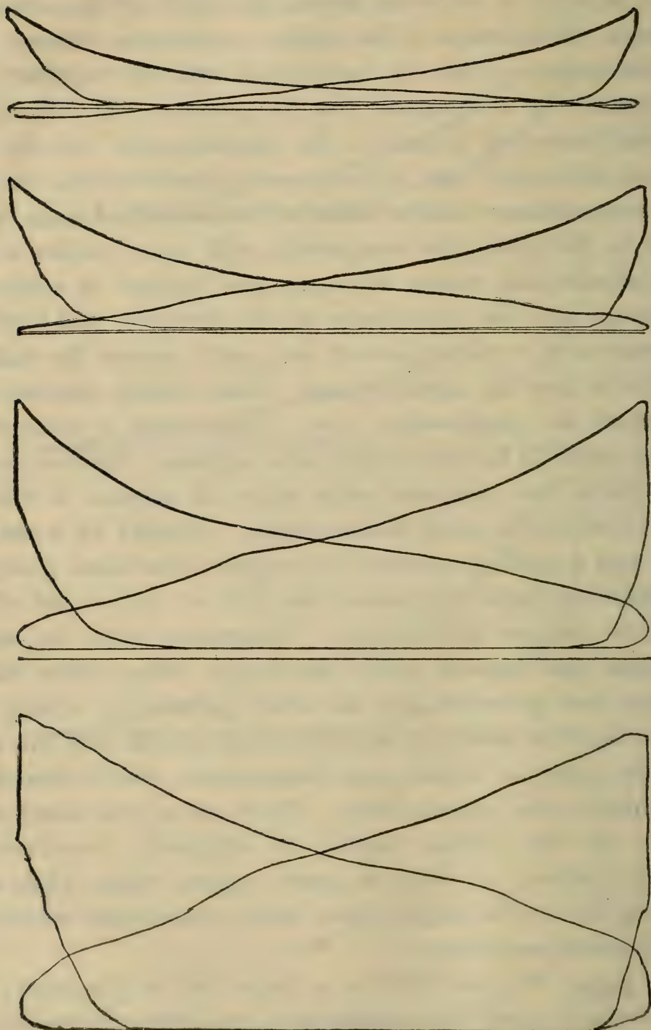


Fig. 246. Diagrams from a Dickson throttling engine.

station. The first diagram was taken late in the afternoon when the engine was started and before any load was thrown on to the machine, and the three succeeding cards were taken at intervals later in the evening as the number of lights increased and the load became heavier. Two or three important points are to be noticed in connection with these diagrams. First, the initial pressure of the steam at the point of admission is very nearly the same in all four cards, the slight variations being due chiefly to a variation in the boiler pressure. Second, the length of the cut-off increases with the load. The compression also becomes later as the cut-off lengthens, and while there is also a change in the points of admission and release, it is not as marked as the changes in cut-off and compression, for reasons that have already been explained.

Taking the cards on page 376, we have four excellent examples of the action of a throttling engine. These cards are from a Dickson engine, taken at the same station and under the same conditions as the Ball engine cards, with the exception that in this case both head and crank-end diagrams were taken on the same cards, while only the head end diagrams from the Ball engine are shown. The two sets of diagrams are well adapted for comparison, because both engines are of the single-valve type, with the valve moved by one eccentric.

The points to be noted are, first, that the points of cut-off are the same, namely at about $\frac{3}{4}$ stroke, in all the throttling cards, and second, that the power of the engine is increased by the action of the governor in opening a throttle valve wider, allowing steam to enter the cylinder at higher pressure.

It was stated at the outset that automatic regulation is the most approved method for regulating the speed of steam engines at the present time. It is generally believed and it is probably true, that automatic engines give better economy than throttling engines and that they regulate a little more closely. It will readily be seen that when the governor of the automatic engine

changes position, it measures out just the quantity of steam that will be required to keep the engine within the speed limits during the following stroke. The effect of this regulation, moreover, is felt at one point in the stroke only — the point of cut-off — so that any change in the governor up to the time when the piston nears the point of cut-off will produce an immediate change in the quantity of steam admitted. In the throttling engine, on the other hand, the regulation is effected during the whole stroke up to the point of cut-off, and the full effect of any change of the governor cannot be felt until the next stroke. With regard to the relative economy of the two types, it should be kept in mind that the throttling engine is generally of cheap construction, has large clearance, a single, unbalanced slide-valve that does duty for both entering and exhaust steam and aside from the throttling feature, is inferior to the average automatic engine. It is reasonable to suppose, therefore, that at least a part of the large steam consumption generally attributed to the throttling engine is due to its inferior design and construction and not to its method of governing.

For example, take the case of the Ball and the Dickson engines, from which cards are shown. They both have a single slide-valve, but the former runs at higher speed than the latter and its valve is balanced, so that for these reasons it would be expected to be a little more economical. We should not expect, however, that a test would show any decided superiority that could be attributed to the method of governing. If we were to compare the average throttling engine with the most approved type of automatic engine, like the Corliss, we should find that the efficiency of the latter was much higher. The gain, however, would be due to a large extent to the small clearance spaces, separate steam and exhaust valves, and other important features of the Corliss engine, rather than to its automatic cut-off. It is not the purpose to discuss here why these features give improved

economy over the single valve, but simply call attention to the fact that they exert an important influence. The exact influence, which the throttling or automatic features exert apart from the general constructive features of the engine is hard to determine. It is known that high-pressure steam is more economical to use than low pressure steam and the automatic engine, which preserves nearly the boiler pressure up to the point of cut-off, gains on this account. On the other hand, it is known that the most economical point of cut-off for a non-condensing engine is about one-third stroke, and when it becomes very much less than this there is a serious drop in the economy. A very short cut-off with high-pressure steam produces so great a variation in the temperature during one stroke of the piston that the cylinder condensation becomes excessive. For very light loads, therefore, it would be better to throttle the steam than to shorten the cut-off.

It is necessary for all engines to have a reserve of power and hence the cut-off of throttling engines must come late in the stroke. If it were early in the stroke, there would not be enough reserve power with the reduction in the pressure of the steam that is necessary with this type. The late cut-off produces poor economy when the load is heavy, because there will then be a high terminal pressure, and a large amount of heat, corresponding to this pressure, will be thrown away. A throttling engine therefore, may be expected to do better at light loads than at heavy ones, and in fact, may do a little better at light loads than the automatic engine. If a throttling engine could be run so as not to vary much from its most economical load, and could be designed to have the good features of the best automatic engines, with the cut-off at an earlier period in the stroke, it would probably be nearly or quite as well as the automatic engine. Under the conditions that they have to run, however, the automatic engine will keep the lead, although, as explained above, its superiority is not due entirely to the automatic feature.

CHAPTER XV.

ECONOMY AND OPERATION OF ENGINES.

Engineers over the country have been discussing whether or not more steam is used when an engine is made to run faster without changing either the cut-off or the back pressure. Some, strange as it may seem, have actually held to the opinion that, since the cut-off is not changed, no more steam is used, and hence, if it were possible to make an engine run faster without changing the cut-off, it would be doing more work than before without any increase in the consumption of steam. Of course, this is wrong. The speed of an engine, almost any engine, may easily be increased without changing the cut-off, and when this is done, the engine will do more work and will use more steam. It is utterly impossible to get something for nothing out of a steam-engine, or out of any engine or appliance. The only way in which a steam-engine can be made to do more work without using more steam is to increase its efficiency. And when everything else is kept the same and the speed only of an engine increased, the efficiency is very slightly increased. The condensation is decreased with an increase of speed, but the decrease would be so slight for most cases that it would hardly be worth considering. When an engine is cutting off at a certain part of the stroke, it uses at every stroke a certain weight of steam which depends upon the initial pressure of the steam, clearance volume of the engine and the point of cut-off. If the engine makes 400 strokes per minute (200 revolutions, if a double acting engine) the weight of steam used will be

400 times the weight used in one stroke; but if the engine be made to make 500 strokes per minute, the weight of steam used per minute will be, neglecting the small difference in condensation, 500 times the weight used in one stroke.

HOW TO INCREASE THE SPEED, OR INCREASE THE POWER OF A CORLISS ENGINE.

There are three ways in which this can be done. Take, for example, a 24″ x 48″ simple Corliss engine making 70 revolutions per minute, the boiler gauge pressure 80 lbs. per square inch, one-quarter cut-off, or cut-off 12 inches from the beginning of the stroke; the mean effective pressure, say about 42 lbs. per sq. in., the governor pulley on the main shaft 10 inches in diameter, the pulley on the governor shaft 7 in. in diameter, and the friction of engine, cylinder clearance, condensation, etc., left entirely out of the question. It is desired to increase the speed of this engine to 80 revolutions per minute, and in this manner increase its horse-power.

First method. — Regardless of piston rod, the area of the piston is 452.4 square inches, nearly. The piston speed of this engine is 560 feet per minute, and its horse-power 322, nearly. Thus: $\frac{452.4 \times 42 \times 560}{33000} = 322$. So that the horse-power of this engine at 70 revolutions per minute is 322, nearly, and this is what the manufacturer's catalogue gives. Now, in order to get 80 revolutions per minute, take the 7-inch pulley off the governor shaft, and put in its place an 8-inch pulley. Thus: 70 : 80 : : 7 : 8. Then, the governor balls will revolve in the same relative plane that they did before, and the cut-off will remain the same; that is, at one-quarter, or 12 in. of the stroke. Thus, 7 : 10 : : 70 : 100. And 8 : 10 : : 80 : 100. So the governor balls make 100 revolutions per minute, both before and after making the change.

Now, with the engine speeded up to 80 revolutions per minute, we get 46 more horse-power. Thus: Piston speed equals 640 feet per minute. Then, $\frac{452.4 \times 42 \times 640}{33000} = 368$ horse-power, nearly. And 368 minus 322 = 46. Now, it would appear that we are getting 46 horse-power more for nothing, but such is not the case. For, $\frac{452.4 \times 12 \times 2 \times 70}{1728} = 439.8+$, or nearly 440 cubic ft. of steam per minute, at 80 lbs. boiler pressure, are required to develop 322 horse-power. And, $\frac{452.4 \times 12 \times 2 \times 80}{1728} = 502.6+$ or nearly 503 cubic ft. of steam per minute, at 80 lbs. boiler pressure, are required to develop 368 horse-power. Then, 503 minus 440 = 63 cubic feet more of steam at 80 lbs. boiler pressure, which means more water evaporated per minute and more coal burned per hour.

Second method. — Retain the same engine speed and the same cut-off, but increase the boiler pressure from 80 to 90 lbs. Then $80 : 90 :: 42 : 47+$, call it 48 lbs. mean effective pressure. Then, $\frac{452.4 \times 48 \times 560}{33000} = 368$ horse-power, nearly, the same as before, and as given in the manufacturer's catalogue. We are now using 440 cubic feet of steam per minute at 90 lbs. pressure, with an increase of 6 lbs. M. E. P.; consequently, more coal per hour must be burned.

Third method. — Retain the same boiler pressure, that is 80 lbs., and weight the governor so as to make the balls revolve in a lower plane in order to give a later cut-off. Thus, $322 : 368 :: \frac{1}{4} : \frac{2}{7}$. That is, the cut-off must take place at about $\frac{2}{7}$ of the stroke instead of at $\frac{1}{4}$. Then, $\frac{1}{4} : \frac{2}{7} :: 42 : 48$. That is the M. E. P. will be 48 lbs. per square inch with a cut-off at $\frac{2}{7}$ of the stroke. Then, $\frac{452.4 \times 48 \times 560}{33000} = 368$ horse-power, the same as

before. But, $\frac{2}{7}$ of $48 = 13\frac{5}{7}$, or 13.71 inches nearly, so that, instead of cutting off at 12 inches with 80 lbs. boiler pressure, we are cutting off at 13.71 inches and using 63 cubic feet more steam per minute. Thus, $\frac{452.4 \text{ x } 13.71 \text{ x } 2 \text{ x } 70}{1728} = 503$, nearly. And, 503 minus $440 = 63$, that is, we must use 63 cubic feet more of steam per minute at 80 lbs. boiler pressure, in order to get 46 more horse-power, which means the evaporation of more water per minute, and the burning of more coal per hour.

HOW TO INCREASE THE HORSE-POWER OF AN ENGINE HAVING A THROTTLING GOVERNOR.

There are three ways in which this can be done, also. We will take, for example, a plain slide-valve engine 10 x 16 inches, making 150 revolutions per minute, with $\frac{9}{16}$ cut-off, and M. E. P. say $31\frac{1}{2}$ lbs. per square inch, with a boiler pressure of 60 lbs. by gauge. The governor pulley on the main shaft 6 inches in diameter, and the pulley on the governor shaft 4 inches in diameter. The horse-power of this engine is about 30. Thus, $\frac{16 \text{ x } 2}{12} = 2\frac{2}{3}$ ft., and $150 \text{ x } 2\frac{2}{3} = 400$ ft., the piston speed. Then, $\frac{10 \text{ x } 10 \text{ x } .7854 \text{ x } 31.5 \text{ x } 400}{33000} = 30$ horse-power, nearly.

It is now desired to run the engine at 180 revolutions per minute in order to develop 6 horse-power more. In order to obtain these results, the governor pulley must be enlarged, so as to make the governor balls revolve in the same plane at 180 revolutions per minute, that they now do at 150 revolutions. Thus, 4 : 6 : : 150 : 225, that is, the governor balls are now making 225 revolutions per minute. And 150 : 180 : : 4 : 4.8. Consequently, the governor pulley must be increased to 4.8 inches in

diameter. Then, 4.8 : 6 : : 180 : 225, that is, the governor balls, after the change, making the same number of revolutions as before. At 180 revolutions per minute, the piston speed is 480 feet per minute. Thus, $\frac{16 \times 2}{12} = 2\frac{2}{3}$. And, $180 \times 2\frac{2}{3} = 480$. Then, $\frac{78.54 \times 31.5 \times 480}{33000} = 36$ horse-power, nearly. It might seem from the above that we are getting 6 horse-power more for nothing; but such is not the case. For, cutting off at $\frac{9}{16}$ is equivalent to cutting off at 9 inches of the stroke.

Then, $\frac{78.54 \times 9 \times 2 \times 150}{1728} = 123$ cubic ft., nearly.

And, $\frac{78.54 \times 9 \times 2 \times 180}{1728} = 147$ cubic feet, nearly. And, 147 minus 123 = 24. So that for 6 horse-power more, we are using 24 cubic feet more of steam per minute, at 31.5 lbs. M. E. P., which means more water evaporated per minute and more coal burned per hour.

If the boiler pressure may be safely increased, we can get 6 horse-power more out of the engine without increasing its speed, by running the boiler pressure up to 75 lbs. by gauge. Thus 75 lbs. boiler pressure would give about 37.8 lbs. M. E. P. with $\frac{9}{16}$ cut-off. Then, $\frac{78.54 \times 37.8 \times 400}{33000} = 36$ horse-power, nearly.

In this case no change should be made in the governor, nor in the speed of the engine. We can also get 6 horse-power more out of this engine by cutting off later, say at $\frac{5}{8}$, in order to get 37.8 lbs. M. E. P. But a later cut-off is not desirable, because it is not economical of steam, and besides, it would require a new valve, new eccentric, or a change in the length of a rocker arm, if not a change of the valve-seat, because the travel of the valve would have to be increased.

HOW TO INCREASE THE HORSE-POWER OF AN ENGINE HAVING A SHAFT GOVERNOR.

Suppose it is desired to increase the speed of the engine from 250 to 275 revolutions per minute, cutting off at $\frac{1}{4}$ stroke. In this case the governor springs should be so adjusted that the throw of the eccentric will be the same at 275 revolutions that it was at 250 revolutions. This will require an increased consumption of steam per minute at the same initial cylinder pressure as before making the change, consequently more fuel will be required. If the speed of the engine is not to be changed, an increase of the horse-power may be obtained by increasing the initial cylinder pressure, if the condition of the boiler will so permit. Or, the initial cylinder pressure may remain unchanged and the governor springs and levers so adjusted as to give a later cut-off, say at $\frac{3}{8}$ or $\frac{7}{16}$ of the stroke, or whatever may be required to offset the increased permanent load, the speed of the engine remaining unchanged. Any one of the changes above described would necessitate an increased consumption of fuel.

HOW TO LINE THE ENGINE WITH A SHAFT PLACED AT A HIGHER OR A LOWER LEVEL.

We will suppose the latter shaft not yet in place, but to be represented by a line tightly drawn. From two points as far apart as practicable, drop plumb lines nearly, but not quite, touching this line. Then by these strain another line parallel with the first, and at the same level as the center line of the engine, and at right angles with this stretch another representing this center line, and extend both each way to permanent walls on which their terminations, when finally located, should be carefully marked, so they can at any time be reset. The problem is to get the latter line exactly at right angles with the former. Everything depends upon the accuracy with which this right

angle is determined. It is done by the method of right-angle triangles. There are two ways of applying this method. In the first, one end of a measuring line is attached to some point of line No. 1, and its other end is taken successively to points on line No. 2 on opposite sides of the intersection, as illustrated in the following figure, in which *A B* is a portion of line No. 1, and *C D* of line No. 2, the direction of which is to be determined. *B F* and *B G* are the same measuring line fixed at *B*, and applied to the line *C D* successively at the points *F* and *G*. The dis-

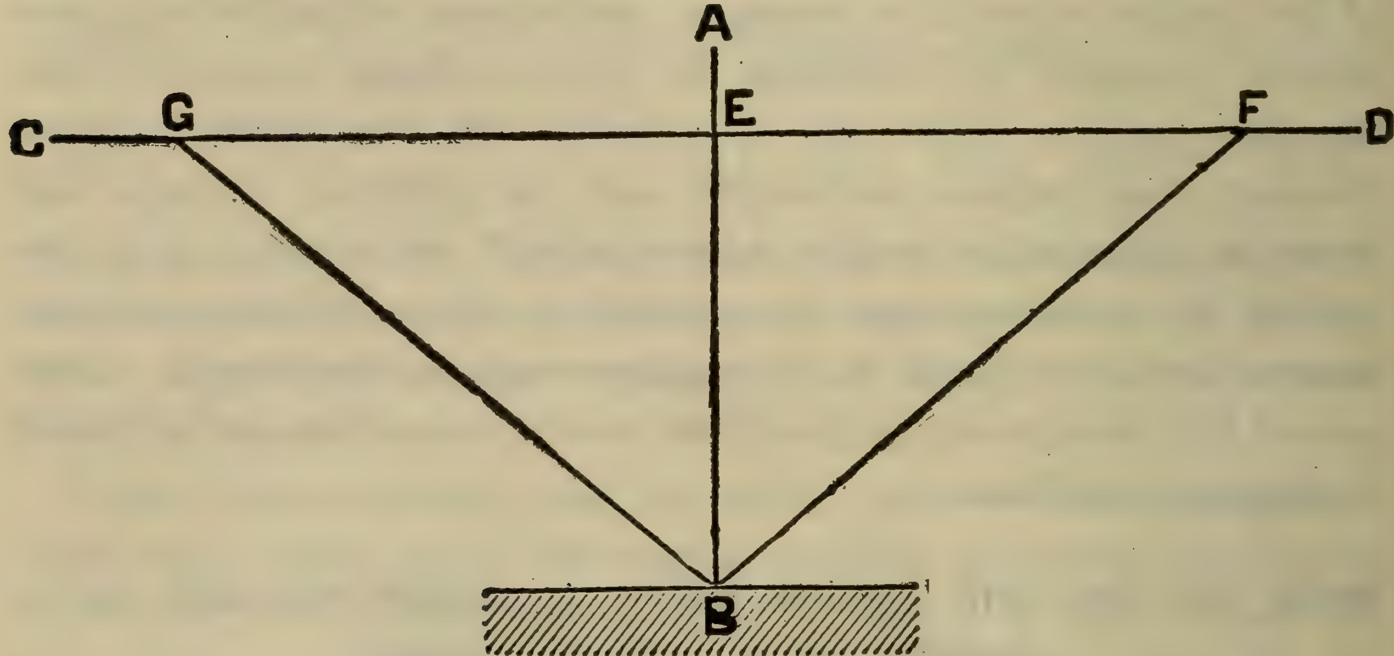


Fig. 247. Lining engine with line shafting.

tances *B F* and *B G* being, therefore, the same, when *E F* is equal to *E G*, the lines *A B* and *C D* are at right angles with each other. In the second, application is made of the law that the square of the hypothenuse of a right-angle triangle is equal to the sum of the squares of the other two sides. Thus $3^2 + 4^2 = 5^2$. So if the above figure $E\ B = 4$, $E\ F = 3$, and $B\ F = 5$, the angle at *E* is a right-angle. Any unit of measure may be used, a foot is generally the convenient one; so any multiple of these numbers may be taken; as, for example, 6, 8 and 10. Respecting the comparative advantages of these two ways, the situation will often determine which is to be preferred. In the former, the diagonal

being the same line, fixed at B and brought successively to the points F and G, its length is immaterial, though generally the longer the better; and the only point to be determined is the equality of $E F$ and $E G$, which may be compared with each other by marks on a rod. In the latter, the proportionate lengths, 3, 4 and 5, or their multiples, must be exactly measured. It is better adapted to places where a floor is laid and the measurements can be transferred by trammels. The result should be verified by repeating the operation on the opposite side of the intersection at E, and when so verified we have, in fact, the first process, without the additional and unnecessary trouble of determining the relative lengths of the lines. Care should be taken when a measuring line is used, to avoid errors from its elasticity. On this account, a rod is often employed. Points on the lines are best marked by tying on a white thread.

HOW TO LINE THE ENGINE WITH A SHAFT TO WHICH IT IS TO BE COUPLED DIRECT.

In this case, it is supposed that the engine bed and the bearings for the shaft are already approximately in position. They are leveled by a parallel straight edge and a spirit level. To line them horizontally, a line must be run through the whole series of bearings and continued to a permanent wall at each end, and its terminating points, when determined, carefully marked, as already directed. A piece of wood is tightly set in each end of each bearing and the surfaces of these are painted white or chalked. Then the middle of each piece being found by the compasses, two fine lines are drawn across it, equally distant from the middle, and having between them a space a little wider than the thickness of the line. The line being tightened, nearly touching the blocks, or, if long, having its sag supported by them, the two marks on each block must be seen, one on each side of the line, with the line of white between.

HOW TO SET A SLIDE VALVE IN A HURRY.

Open the cylinder cocks; then open the throttle slightly, so as to admit a small amount of steam to the steam-chest. Roll the eccentric forward in the direction the engine runs, until steam escapes from the cylinder cock at the end where the valve should begin to open. Now screw the eccentric fast to the shaft. Roll the crank to the next center and ascertain if steam escapes at the same point, at the opposite end of the cylinder. If so, ring the bell and go ahead. The valve gear can be run until an opportunity occurs to remove cover from steam-chest and examine the valve.

DO YOU DO THESE THINGS?

A writer in a magazine asks and answers the following pertinent questions: —

Do you take a squirt-can in one hand and project a stream of oil as far as you can throw it, in order to save going to the oil hole itself?

If you do, don't do it any more; willful waste is downright robbery.

Do you use an oil can at all for oiling, except on emergency, or for the moment?

If you do, don't do it any more, for much better lubrication can be had by automatic apparatus.

Do you keep an old tin coffee-pot full of suet on the steam-chest, and every time you have nothing else to do, pour a dipperful into the steam-chest?

If you do, stop it and get a sight-feed cup, which will save you the labor of slushing the cylinder and save the cylinder and valve-seats, the piston and follower, and all other places touched by the grease.

Do you feed the boiler until the water is out of sight in the glass, then shut off the feed, put in a big fire and sit down in a dark corner with a four-horse brier pipe and smoke, until you happen to think that maybe the water is low?

If you do these things you should notify the coroner that some day his services will be needed, but it is better to cease the practice mentioned before the coroner comes.

Do you stop leaks about the boiler as fast as they occur, or do you wait until the places sound like a snake's den before you stir?

If you do, you waste heat, which is the same word as money, only differently spelled. Every jet of hot water leaking from a steam boiler is just so much money thrown away, and if it was your money you would be bankrupt in a short time, in some boiler rooms.

Do you take a screw wrench and yank away at a bolt or nut under steam pressure?

If you do, there will come a time, sooner or later, when you will do so once too often, and either kill yourself or some one else. Bolts and nuts are liable to strip or break if tampered with under pressure, and they never tell any one beforehand when they are going to do it.

Do you attempt to stop pounding in the engine by laying for the crank-pin as it comes round, and trying to hit the key once in a while?

If you do, ask the strap and neck of the connecting-rod how he likes it, when you don't hit the key and do hit the oil cup?

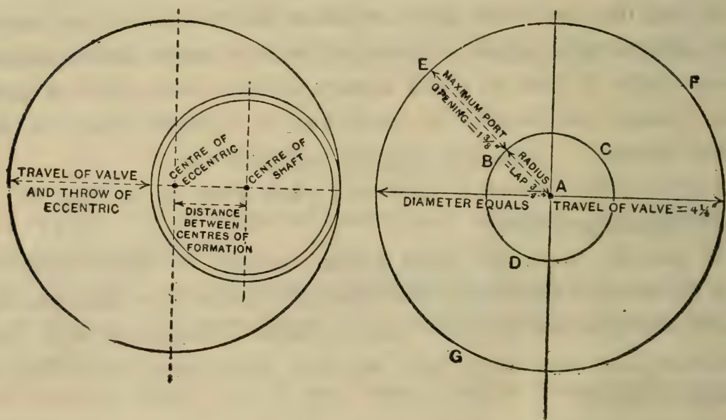
Do you pack the piston by taking it out of the cylinder, laying it on the floor, setting out the rings, and then when the piston will not go into the cylinder, try to batter it in with a four-foot stick of cordwood?

If you do, you should reform, and pack the piston in the cylinder where it belongs, being sure to get it central by measuring from the lathe center in the end of the piston rod.

Do you put a new turn of packing on top of the old, hard-burned stuff when the piston rod leaks steam?

If you do, you will have a scored piston rod and broken gland bolts some day. Packing under heat and pressure gets so hard that it cuts like a file when left in the stuffing-box, and as one begins to leak all the old stuff should be pulled out and new put in its place.

THE TRAVEL OF A SLIDE VALVE.



Figs. 248 and 249. The throw of the eccentric.

The travel of a slide valve is found as follows: The maximum port opening at the head end, plus the maximum port opening at the crank end, plus the lap at the head end, plus the lap at the crank end. Therefore, $1\frac{3}{8}'' + 1\frac{3}{8}'' + \frac{3}{4}'' + \frac{3}{4}'' = 4\frac{1}{4}''$, the required travel of valve. Incidentally, it may be well to mention that the travel of a valve may also be obtained from the eccentric, by subtracting the thin part of the eccentric from the thick part as per Fig. 248, or again, by taking twice the distance between the center of rotation and center of the eccentric. This distance on the eccentric is the end valve travel, and is termed the "throw" of the eccentric. In the above question, the travel may also be

found by the aid of the diagram, Fig. 249, which is explained as follows: From the center A, with a radius of $\frac{3}{4}$ inch (lap), describe a circle $B\ C\ D$. From any point in the circumference, say B, lay off the distance $B\ E$ equal to the maximum port opening, $1\frac{3}{8}''$; from the center A, with a radius $A\ E$, describe the circle $E\ F\ G$; the diameter of the circle $E\ F\ G$ is equal to the travel of the valve, which is $4\frac{1}{4}''$. Let the reader try this with another set of figures, to prove the correctness of the diagram.

LOSS OF HEAT FROM UNCOVERED STEAM PIPES.

The following table shows the loss of heat through naked steam pipes, wrought iron, of standard sizes. The best covering for a steam pipe is hair felt from one to two inches thick, depending on the diameter of the pipe, say one inch thick for pipe from 1 to 4 inches in diameter, and two inches or more for larger pipes. Such covering will save at least 96 per cent. Cheaper coverings will save from 75 to 90 per cent. The chief value of the table is as an aid in estimating the saving that can be made by covering the pipe. The money loss by naked pipe being known, the saving can be estimated and the cost of the covering will decide its value as an investment.

TABLE OF MONEY LOSS FROM 100 FEET OF NAKED STEAM PIPE, FOR ONE YEAR OF 3000 WORKING HOURS.

Nominal diameter of pipe in inches.	STEAM PRESSURES.					
	50 lbs.	60 lbs.	70 lbs.	80 lbs.	90 lbs.	100 lbs.
1	$13.15	$13.70	$14.20	$14.66	$15.08	$15.47
$1\frac{1}{4}$	16.58	17.29	17.92	18.49	19.02	19.51
$1\frac{1}{2}$	18.98	19.78	20.51	21.17	21.77	22.33
2	23.72	24.73	25.63	26.45	27.21	27.91
$2\frac{1}{2}$	28.72	29.94	31.03	32.03	32.94	33.79
3	34.97	36.45	37.78	38.99	40.10	41.14
4	44.96	46.86	48.57	50.13	51.56	52.89
5	55.57	57.92	60.04	61.96	63.73	65.38
6	66.27	69.08	71.60	73.89	76.01	77.96

RULES AND PROBLEMS APPERTAINING TO THE STEAM ENGINE.

To find the H. P. of a simple non-condensing engine: —

Rule. — Multiply the net area of the piston in square inches, by the mean effective pressure in pounds per square inch, and by the velocity of the piston in feet per minute, and divide the last product by 33,000. The quotient will be the gross H. P. Subtract from this from ten to twenty per cent for friction in the engine itself, and the remainder will be the delivered H. P.

Example. — The area of the piston is 500 sqr. ins. Half the area of the piston-rod is 5 sqr. ins. The M. E. P. is 50 lbs. per sqr. in. The stroke is 3 feet, and the revolutions per minute 125. The friction is 10 per cent. What is the delivered H. P. of the engine? Ans. 506.25 H. P.

Operation. — 3 ft. $\times$ 2 = 6 ft. twice the stroke.

Then, 500 — 5 = 495 sqr. ins. net area of piston.

And, 125 $\times$ 6 = 750 ft. the piston speed per minute.

And, $\dfrac{495 \times 50 \times 750}{33,000} = 562.5.$

Then, 562.5 $\times$ .90 = 506.25. The delivered H. P.

For a condensing engine: — Add the vacuum to the M. E. P. and proceed as above.

The M. E. P. is the average pressure in the cylinder, less the back pressure.

To find the H. P. of a compound noncondensing engine: —

The usual method of calculating the H. P. of a multiple cylinder engine is to assume that all the work is done in the low pressure cylinder alone, and that such a M. E. P. is obtained in that cylinder as will give the same H. P. as is given by the whole engine.

Rule. — Find the ratio of areas of the high and low pressure cylinders, — when of the same stroke, as they usually are, — and

multiply it by the number of expansions in the high pressure cylinder, for the total number of expansions in both cylinders. Find the hyperbolic logarithm corresponding to this result and add 1 to it, and divide the sum by the total number of expansions. Multiply this result by the absolute steam pressure, and subtract the back pressure. Subtract again the loss in pressure between cylinders, and the remainder will be the M. E. P. Then multiply the net area of the low pressure cylinder by this M. E. P. and by the piston speed in feet per minute and divide by 33,000. Deduct the friction in the engine itself and the remainder will be the delivered H. P.

Example. — Given a tandem compound engine with cylinders 20″ and 32″ diameter, and 4 feet stroke, making 75 revolutions per minute, boiler gauge pressure 125 lbs. per sqr. in., $\frac{1}{4}$ cut-off in high pressure cylinder, back pressure $15\frac{1}{4}$ lbs. per sqr. in., drop in pressure between cylinders 15 per cent, and friction in engine 10 per cent. What is the H. P. delivered of this engine? Ans. 338.4 H. P.

Operation. — Neglecting the areas of the piston rods, we have: —

$20 \times 20 \times .7854 = 314.16$ sqr. ins. area of high pressure cylinder.

And, $32 \times 32 \times .7854 = 804.2$ sqr. ins. area of low pressure cylinder.

Then, $804.2 \div 314.16 = 2.56 =$ the ratio between cylinders.

And, $2.56 \times 4 = 10.24 =$ the total number of expansions. The hyperbolic logarithm of $10.24 = 2.328$. (See table on p. 397.)

And, $1 + 2.328 = 3.328$.

Then, $3.328 \div 10.24 = .325$.

Also, $125 + 15 = 140$ lbs., the absolute pressure.

And, $.325 \times 140 = 45.5$ lbs., forward pressure.

And, $45.5 - 15.25 = 30.25$ lbs., the M. E. P.

And, $30.25 \times .85 = 25.7$ lbs. = the M. E. P. less the "drop."

Then, $\frac{804.2 \times 25.7 \times 8 \times 75}{33,000} = 376$ H. P. nearly.

And, $376 \times .90 = 338.4$ H. P. delivered.

For a compound condensing engine, proceed as above, except that the condenser pressure, due to impaired vacuum, only should be subtracted from the forward pressure.

To find the linear expansion of a wrought-iron pipe or bar: —

Rule. — Multiply the length of the pipe or bar *in inches* by the increase in temperature, and by the constant number .0007, and divide the last product by 100.

Example. — Given a 6-inch wrought-iron pipe 75 feet long. Steam pressure 150 lbs. by gauge. Temperature of pipe when put up 60 degs. Fah. What is its linear expansion? Ans. 2 ins. nearly.

Operation. — The diameter of the pipe is immaterial.

Then, 150 lbs. pressure = 366 degs.

And, $366 - 60 = 306$ degs.

Also, $75 \times 12 = 900$ inches length of pipe.

Then, $\frac{306 \times 900 \times .0007}{100} = 1.9278$ inch.

For copper, use the constant number .0009; for brass, use .00107; for fire-brick, use .0003, and proceed as above.

To find the proper diameter of steam pipe for an engine: —

The velocity of steam flowing to an engine should not exceed 6,000 feet per minute.

Rule. — Multiply the area of the piston in square inches by the piston speed in feet per minute, and divide by 6,000; and divide again by .7854, and extract the square root for the diameter of the pipe and take the nearest commercial size.

Example.—Given a 20″ × 48″ Corliss engine making 72 revolutions per minute. What should be the diameter of its steam pipe? Ans. 6 inches.

Operation. — $20 \times 20 \times .7854 = 314.16$ sqr. ins.

And, $\frac{48'' \times 2 \times 72}{12} = 576$ ft. the piston speed.

And. $\frac{314.16 \times 576}{6,000} = 30.15.$

Then, $\sqrt{\frac{30.15}{.7854}} = 6.1''$. Take 6″ pipe.

To find the water consumption of a steam engine: —

The most reliable method for determining this, is to make an evaporation test, that is, to measure the water fed to the boiler in a given time and delivered to the engine in the form of steam. But as this method entails considerable trouble and expense, it is frequently figured from indicator diagrams. This plan, however, does not insure correct results, because the amount of water accounted for by the indicator is considerably less than it should be owing to cylinder condensation and leakage, so that it might be possible that only 80 per cent of the water passing through the cylinder would be accounted for by the indicator. But the calculation, used in connection with an evaporation test, will reveal the extent of the losses caused by cylinder condensation and leakage, by deducting the amount of water found by computation from the amount of water fed to the boiler while making an evaporation test.

Rule. — Divide the constant number 859,375 by the M. E. P. of any indicator card, and divide this quotient by the volume of its total terminal pressure, the result will be the theoretical consumption in pounds of water per horse power per hour.

The constant number 859,375 is found as follows: —

Compute the size of an engine that will give just one horsepower at one pound M. E. P. per square inch, thus:

Area of piston equals 412.5 sqr. inches.

Stroke equals 4 feet, and revolutions per minute equal 10.

Then, the piston speed is ($4 \times 2 \times 10$) 80 feet per minute.

And, $\frac{412.5 \times 1 \times 80}{33,000} = 1.$

To find how much water it would take to run this engine *one hour*, allowing $62\frac{1}{2}$ lbs. to the cubic foot of water, proceed as follows: —

Twice the stroke equals 96 inches.

Then, $\frac{412.5 \times 96}{1728}$ equals 22.91666 cubic feet for one revolution.

And, 22.91666×10 equals 229.1666 cubic feet for 10 revolutions, or for *one minute*.

Then, $229.1666 \times 60 \times 62\frac{1}{2}$ equals 859,375 lbs. of water used per hour.

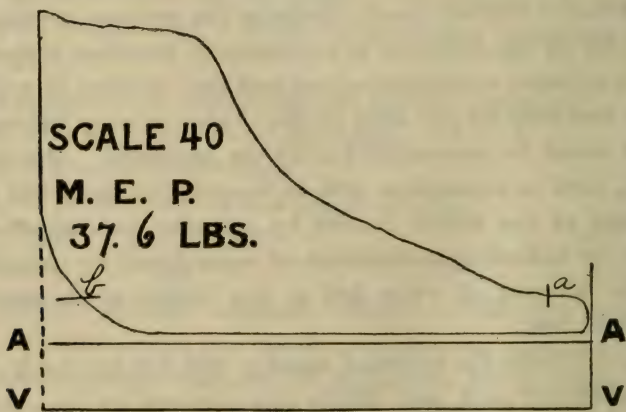


Fig. 250. Finding steam consumption from the diagram.

Fig. 250 is not an actual indicator card, but answers to illustrate the rule.

AA is the atmospheric line, and from *A* to *A* is the whole stroke.

VV is the vacuum line.

Points (*a*) and (*b*) are equally distant from the vacuum line. The point (*a*) is taken at or very near the point of release.

Example. — From the indicator card Fig. 243 compute the water consumption, the M. E. P. being 37.6 lbs. per square inch, the scale of spring used in the indicator being 40, the distance from point (a) to point (b) being 3.03 inches, the stroke AA being 3.45 inches, and the pressure at point (a) being 25 lbs. per sqr. inch absolute. Ans. 20.14 lbs.

Operation. — $859{,}375 \div 37.6 = 22{,}855.7$.

Now, the absolute pressure at point (a) is 25 lbs., and steam tables give 996 as the volume of steam at this pressure, that is, steam at this pressure has 996 times the bulk of the water from which it was generated.

Then, $22{,}855.7 \div 996 = 22.94$ lbs. of water. But as the period of consumption is represented by (b) (a), AA being the whole stroke, the following correction is required: The distance from point (a) to point (b) is 3.03 ins. Then, $22.94 \times 3.03 = 69.5080$. And the whole stroke or length of line AA is 3.45 ins.

Then, $69.5080 \div 3.45 = 20.14$ lbs. of water per indicated horse power per hour.

Boiler Feed or Pressure Pumps.

SIZES AND CAPACITIES.

Steam Cylinder. Inches.	Water Cylinder. Inche	Stroke. Inches.	Gallons per Stroke.	Capacity per minute at ordinary speed.	Steam pipe. Inches.	Exhaust pipe. Inches.	Suction pipe. Inches.	Delivery pipe. Inches.	Floor Space Required. Ins.	Horse-power of boiler best adapted for.
2½	1½	3	.023	150 Strokes. 3½ gals.	¼	⅜	½	⅜	17x .5	
3	1¾	3	.031	150 " 4¾ "	¼	⅜	¾	½	18x 5	
3¼	2	4	.05	150 " 7½ "	½	¾	1¼	1	26x 6	25
3½	2¼	4	.07	150 " 10½ "	½	¾	1¼	1	28x 7	40
4	2½	5	.11	150 " 16½ "	½	¾	1¼	1	31x 8	60
5	3¼	7	.25	125 " 31 "	¾	1	2	1½	44x13	90
5½	3¾	7	.35	125 " 42 "	¾	1	2	1½	45x14	130
7	4	7	.39	125 " 49 "	1	1¼	2½	2	45x14	160
7	4½	10	.69	100 " 69 "	1	1¼	3	2½	55x16	200
7½	5	10	.85	100 " 85 "	1	1¼	3	2½	55x16	250
8	5	12	1.02	100 " 102 "	1	1¼	4	4	67x19	300
10	6	12	1.47	100 " 147 "	1¼	1½	4	4	67x19	400
12	7	12	2.00	100 " 200 "	2	2½	5	5	67x20	600
14	8	12	2.61	100 " 261 "	2	2½	5	5	67x20	
16	10	16	5.44	75 " 408 "	2½	3	6	6	80x22	
18	12	24	11.75	50 " 588 "	3½	4	8	6	110x27	
20	14	24	16.00	50 " 800 "	3½	4	10	8	111x29	

CHAPTER XVI.

THE STEAM BOILER.

THE FORCE OF STEAM AND WHERE IT COMES FROM.

If water be heated it will expand somewhat, and will finally burst forth into vapor. The vapor will expand enormously, and naturally occupy more space than the water from which it is formed. A cubic inch of water will make a cubic foot of steam; that is, the water has been expanded by heat to seventeen hundred times its original bulk. The steam is very elastic; the water was not. When we say that a cubic inch of water will form a cubic foot of steam, we mean that it will do so when the steam is allowed to rise naturally from the water without any confinement. If the steam is confined, as it would be in a boiler, it could not expand, and consequently would not. If the steam is allowed to rise into the atmosphere from an open vessel, the pressure of the steam would be precisely the same as the pressure of the atmosphere, that pressure being about fifteen pounds to the square inch. An ordinary steam gauge only takes notice of the pressure above the atmospheric pressure. When the hand of the steam gauge stands at zero, it indicates that there is no pressure above the ordinary pressure of the atmosphere. An ordinary steam gauge not connected with anything has the atmosphere acting upon it in both directions, the same as the atmosphere acts upon everything when it can reach both sides. If the air be pumped out of the steam gauge, the atmosphere will then act upon one side, and the hand will move backward until it stands at fifteen points less than zero. In this condition the steam gauge indicates the absolute zero of pressure. If now the air be allowed to re-enter where it was pumped out, it will begin to exert its pressure upon the steam

gauge, and the hand will move forward; when the full air pressure is on, the gauge hand will stand at its usual zero.

To go into this matter in order that it may be understood that the real pressure of steam is always fifteen pounds greater than ordinary steam gauges indicate. In all of the finer calculations relating to the action of steam, its total pressure must be known, and this total pressure is to be counted from the absolute zero. The real pressure of steam is always the steam gauge pressure, plus fifteen pounds. When a steam gauge shows fifty pounds, the steam really has a pressure of sixty-five pounds. The fifteen pounds of this pressure is nullified by the atmospheric pressure, and the steam gauge shows us our useful pressure. As before stated, a cubic inch of water will make a cubic foot of steam at atmospheric pressure; that is, fifteen pounds to the square inch, abolute pressure, or zero by the steam gauge. If this cubic inch of water was made into steam in a boiler holding just a cubic foot, the steam gauge would show zero. If the boiler was only large enough to hold half a cubic foot, the steam would all be in the boiler, and being confined in half its natural space, it would have double pressure. It would have an absolute pressure of thirty pounds to the square inch, and the steam gauge would indicate fifteen pounds. If this steam was then allowed to pass into a chamber holding a cubic foot, the steam would expand until it filled the chamber, and its pressure would go down again to fifteen pounds absolute. In short, the pressure is in reverse proportion to the amount of space it occupies. The pressure of steam may be doubled by compressing the steam into one-half its former volume, and so on. After water is turned into steam, the steam may be made hotter, but it is not very much expanded. The pressure of steam is increased by forcing more steam into the space occupied. If a boiler contains steam at 50 lbs. pressure, we may increase the pressure by adding more steam, and thus compressing all the

steam that the boiler contains. In the ordinary operation of a steam boiler, the fire turns the water into steam and the more steam there is made and confined, the greater the pressure will be. If the steam is constantly flowing out of the boiler into an engine, the pressure in the boiler must be kept up by continually making new steam to take the place of that drawn off. If we make steam as fast as it is drawn off, and no faster, the pressure will remain the same. If we make steam faster than the engine draws it off, the pressure will rise, and if it is drawn off faster than we make it, the pressure will go down.

The pressure of the steam is due to its desire to expand into a larger body, and it acts outwardly in every direction against everything upon which it presses. If we crowd 600 cu. ft. of steam in a boiler, which will only hold 100 cu. ft., the steam will be held compressed into one-sixth its natural bulk, and will thus have a pressure of 90 lbs., and the steam gauge will show 75 lbs. If a hole 1 in. square be cut in the boiler, and a weight of 75 lbs. be laid over the hole, the steam will just lift the weight. If the atmospheric pressure could be removed from one sq. in. of the top of the weight, the steam would then be capable of lifting a 90 lb. weight. The force which this steam will exert to lift a weight, or any similar thing against which it acts, will equal the pressure per square inch multiplied by the number of square inches which the steam acts upon. It will thus be readily understood that if we lead a pipe from the boiler and fit a piston in the pipe, the steam will tend to force this piston out of the pipe.

THE ENERGY STORED IN STEAM BOILERS.

A steam boiler is not only an apparatus by means of which the potential energy of chemical affinity is rendered actual and available, but it is also a storage reservoir, or a magazine, in which a quantity of such energy is temporarily held; and this quantity,

always enormous, is directly proportional to the weight of water and of steam which the boiler at the time contains. The energy of gunpowder is somewhat variable, but a cubic foot of heated water under a pressure of 60 or 70 lbs. per square inch, has about the same energy as one pound of gunpowder; at a low red heat, it has about forty times this amount of energy.

The letters B. T. U. are the initial letters of the words British Thermal Unit, and are used as abbreviations of those words. The British Thermal Unit is the unit of heat used in this country and England, and may be said to be the amount of heat required to raise the temperature of one pound of pure water from 39 to 40 degrees Fahr. It is often necessary to distinguish between B. T. U. used in this country and the French thermal unit used in France and most of the countries of Europe. The French thermal unit is called the calorie, and is the heat required to raise the temperature of one kilogram of water one degree centigrade.

Safety at high pressure depends entirely upon the design, material, and workmanship, and it is a question that may be regarded as settled long since, that a steam boiler properly constructed and designed for a working pressure of 150 pounds is as safe as a properly constructed boiler designed for eighty pounds, with the chances in favor of the high pressure, for the reason that less care is taken in selecting boilers for the ordinary pressure, as anything in the shape of a boiler is regarded, by careless people, as good enough for the lower pressures, with which they have become so familiar as to become almost too careless.

SPECIAL HIGH PRESSURE BOILERS.

The extending use of compound steam engines, which make necessary the employment of high steam pressures, calls for steam boilers specially designed to successfully operate under working pressures ranging from 100 to 160 pounds. These boilers must be safe and economical and of such construction as to afford

access for examination and repair, moderate in first cost and maintenance and of simplest possible form. Fortunately, the controlling conditions are not difficult to meet, and there are several well-tried and approved types of steam boilers from which to make a selection, choice being governed by the space at disposal, arrangement of plant, kind of fuel and other circumstances.

TYPES OF BOILERS.

Four types that are very succesfully used, and they represent good practice for high pressure work, being respectively the Horizontal Tubular, and Vertical Fire Box Tubular Boilers. The Fire Box Locomotive Tubular Boiler may safely be added to this list and gives most excellent results.

THE WATER TUBE BOILER.

Steam boilers must be designed with reference to the pressure of steam to be carried, and when so designed and constructed are quite as safe at one pressure as another, preference being given to the type that is simplest in form and the least liable to destruction, not so much by reason of the pressure carried as by failure to provide for the strains of expansion and contraction within itself.

HORSE POWER OF BOILERS.

In determining the proper size or evaporating capacity of a boiler to supply steam for a given purpose, it is necessary to consider the number of pounds of dry steam actually required per hour at the stated pressure. The standard horse power rating for any steam boiler is $34\frac{1}{2}$ pounds of water evaporated (made into steam) from feed water at 212°, per hour. The total pounds steam required for any purpose per hour on this basis divided by $34\frac{1}{2}$ will give the standard boiler horse power required. Manu-

facturers of steam boilers sometimes rate the horse power of their boilers by so many square feet of heating surface per horse power; 8 to 15 sq. ft. of heating surface, they figure, equals one horse power. This rating does not represent the actual capacity of the steam boiler, the only safe guide being the evaporative performance in pounds of steam from water at 212° to steam at 212°. Some boilers will evaporate this with 8 sq. ft., some requiring from 15 to 18 sq. ft., hence, the absurdity of rating horse power of boilers of unlike construction by the square feet of heating surface. But as the practice is an old one in the case of the well-known tubular boiler, so deservedly popular and used more than any other kind, good practice is to allow approximately as follows: —

Allow for each Horse Power —

Steam for Heating, etc.	15 sq. ft. heating surface.
For Plain Throttle Engine, . . .	15 " " "
For Simple Corliss Engine . . .	12 " " "
For Compound Corliss Condensing .	10 " " "

Hence, a boiler for heating purposes or furnishing steam for —

Plain Slide engine with 1,500 sq. ft. surface, equals .	100 H. P.
For Simple Corliss Engine, same boiler " .	125 H. P.
For Compound Condensing Engine " .	150 H. P.

The best method is to compare boilers by their evaporative efficiency and not by heating surface.

The following is an approximate consumption of steam per indicated horse power per hour for engine: —

Plain Slide Engine	60 to 70 pounds.
High Speed Automatic Engine	30 to 50 "
Simple Corliss Engine	25 to 35 "
Compound Corliss Engine	15 to 20 "
Triple Expansion Engine	13 to 17 "

depending upon the horse power, steam pressure, condition of engine, load, etc.

Each pound of first-class steam coal consumed under a well-proportioned steam boiler, well managed, should evaporate 10 pounds of water at 212° into dry steam at 212°. The average boiler throughout the country, with ordinary fuel and management, ranges from 5 to 8 pounds steam per pound of coal, and it would scarcely be safe to make fuel guarantees per horse power of engine without a counter guarantee on the part of the purchaser, when his old boiler is used, that the fuel economy is based on an evaporative efficiency of a given weight of water evaporated per pound of coal per hour in his boiler. The usual practice is to ignore the boiler altogether and guarantee pounds of steam per indicated horse power per hour used by the engine. This affords an exact method and is not hampered by unknown conditions, and places all tests on an equal or comparative basis.

THE RATING OF BOILERS.

It is considered usually advisable to assume a set of practically attainable conditions in average good practice, and to take the power so obtainable as the measure of the power of the boiler in commercial and engineering transactions. The unit generally assumed has been usually the weight of steam demanded per horse power per hour by a fairly good steam engine. In the time of Watt, one cubic foot of water per hour was thought fair; at the middle of the present century, ten pounds of coal was a usual figure, and five pounds, commonly equivalent to about 40 lbs. of feed water evaporated, was allowed the best engines. After the introduction of the modern forms of engine, this last figure was reduced 25 per cent, and the most recent improvements have still further lessened the consumption of fuel and of steam. By general consent the unit has now become thirty pounds of dry steam per

horse power per hour, which represents the performance of non-condensing engines. Large engines, with condensers and compound cylinders, will do still better. A committee of the American Society of Mechanical Engineers recommended thirty pounds as the unit of boiler power, and this is now generally accepted. They advised that the commercial horse-power be taken as an evaporation of 30 lbs. of water per hour from a feed water temperature of 100° Fahr. into steam at 70 lbs. gauge pressure, which may be considered equal to $34\frac{1}{2}$ lbs. of water evaporation, that is, $34\frac{1}{2}$ lbs. of water evaporated from a feed water temperature of 212° Fahr. into steam at the same temperature. This standard is equal to 33,305 British thermal units per hour. A boiler rated at any stated power should be capable of developing that power with easy firing, moderate draught and ordinary fuel, while exhibiting good economy, and at least one-third more than its rated power to meet emergencies.

WORKING CAPACITY OF BOILERS.

The capacity or horse-power of a boiler, as rated for purposes of the trade, is commonly based upon the extent of heating surface which it contains. The ordinary rating was for a long time 15 sq. ft. of surface per horse-power. At the present time most of the stationary boilers are sold on the basis of from 10 to 12 sq. ft. per horse-power, the power referred to being the unit of 30 lbs. evaporation per hour. This method of rating is arbitrary, inasmuch as it is independent of any condition pertaining to the practical work of the boiler. The fact that 10 or 12 sq. ft. of surface is sold for one horse-power is no guarantee that this extent of surface will have a capacity of one horse-power when the boiler is installed and set to work. The boiler in service and the boiler in the shop are two entirely different things, and where one passes to the other, the trade rating disappears. New

conditions, such as draft, grate surface, kind of fuel and management, then take effect, and these have a controlling influence upon the working capacity. The working power may be found to be much less than the arbitrary rate, or it may be a much larger quantity; all depending upon the surrounding conditions. attention is had to this subject, because it is important in some cases to have a clearer understanding as to what is the working capacity of a boiler. Suppose a boiler manufacturer enters into an agreement to install a boiler, which will have a capacity of 100 horse-power. Suppose that on account of poor draft, low grade of fuel, or unfavorable surroundings, all of which are known beforehand, the boiler develops the power named only with the most careful handling. Is the working capacity, under the circumstances, 100 horse-power? Assuredly not, for the purchaser could not depend upon it in ordinary running for that amount of power. Yet the builder may claim that he has fulfilled his contract.

The former boiler test committee of the American Society of Mechanical Engineers established a working rate for boiler capacity which meets such cases in a definite and satisfactory manner. They realized that for the purpose of good work, a boiler should be capable of developing its capacity with a moderate draft and easy firing; and that it should be capable of doing one-third more in cases of emergency. In other words, a boiler which is sold for 100 horse-power should develop $133\frac{1}{3}$ horse-power under conditions giving a maximum capacity. In the instance cited above, the boiler should have been capable of giving 100 horse-power with such ease that there would be a reserve of $33\frac{1}{3}$ horse-power available when urged to this extra power. According to this rule, the capacity of a boiler in a working plant would be found by determining how much water it can evaporate under conditions which will give its maximum capacity; that is, with wide open damper, with the maximum draft available and with the best con-

ditions as to the handling of the fire, and in this way ascertain the maximum power available under these circumstances. Having found this maximum quantity, the working capacity or the rated power would be determined by deducting from the maximum 25 per cent. This rule, it will be seen, does not take into account the extent of the heating surface or the trade rating, but it deals solely with the capabilities of the boiler under the conditions which pertain to its work.

CODE OF RULES FOR BOILER TESTS.

Starting and stopping a test. — A test should last at least ten hours of continuous running, and twenty-four hours whenever practicable. The conditions of the boiler and furnace in all respects should be, as nearly as possible, the same at the end as at the beginning of the test. The steam pressure should be the same; the water level the same; the fire upon the grates should be the same in quantity and condition; and the walls, flues, etc., should be of the same temperature. To secure as near an approximation to exact conformity as possible in conditions of the fire and in the temperature of the walls and flues, the following method of starting and stopping a test should be adopted: —

Standard method. — Steam being raised to the working pressure, remove rapidly all the fire from the grate, close the damper, clean the ash-pit, and, as quickly as possible, start a new fire with weighed wood and coal, noting the time of starting the test and the height of the water level while the water is in a quiescent state, just before lighting the fire. At the end of the test, remove the whole fire, clean the grates and ash-pit, and note the water-level when the water is in a quiescent state; record the time of hauling the fire as the end of the test. The water-level should be as nearly as possible the same as at the beginning of the test. If it is not the same, a correction should be made by computa-

tion, and not by operating pump after test is completed. It will generally be necessary to regulate the discharge of steam from the boiler tested by means of the stop-valve for a time while fires are being hauled at the beginning and at the end of the test, in order to keep the steam pressure in the boiler at those times up to the average during the test.

Alternate method. — Instead of the standard method above described, the following may be employed where local conditions render it necessary: At the regular time for slicing and cleaning fires have them burned rather low, as is usual before cleaning, and then thoroughly cleaned; note the amount of coal left on the grate as nearly as it can be estimated; note the pressure of steam and the height of the water-level — which should be at the medium height to be carried throughout the test — at the same time; and note this time as the time for starting the test. Fresh coal which has been weighed, should now be fired. The ash-pits should be thoroughly cleaned at once before starting. Before the end of the test the fires should be burned low, just as before the start, and the fires cleaned in such a manner as to leave the same amount of fire, and in the same condition, on the grates as on the start. The water-level and steam pressure should be brought to the same point as at the start, and the time of the ending of the test should be noted just before fresh coal is fired.

DURING THE TEST.

Keep the conditions uniform. — The boiler should be run continuously without stopping for meal times, or for rise or fall of pressure of steam due to change of demand for steam. The draught being adjusted to the rate of evaporation or combustion desired before the test is begun, it should be retained constant during the test by means of the damper. If the boiler is not connected to the same steam pipe with other boilers, an extra outlet

for steam with valve in same should be provided, so that in case the pressure should rise to that at which the safety valve is set, it may be reduced to the desired point by opening the extra outlet, without checking the fire. If the boiler is connected to a main steam pipe with other boilers, the safety valve on the boiler being tested should be set a few pounds higher than those of the other boilers, so that in case of a rise in the pressure the other boilers may blow off, and the pressure be reduced by closing their dampers, allowing the damper of the boiler being tested to remain open, and firing as usual. All the conditions should be kept as nearly uniform as possible, such as force of draught, pressure of steam and height of water. The time of cleaning the fires will depend upon the character of the fuel, the rapidity of combustion and the kind of grates. When very good coal is used and the combustion not too rapid, a ten-hour test may be run without any cleaning of the grates, other than just before the beginning and just before the end of the test. But in case the grates have to be cleaned during the test, the intervals between one cleaning and another should be uniform.

Keeping the records. — The coal should be weighed and delivered to the firemen in equal portions, each sufficient for about one hour's run, and a fresh portion should not be delivered until the previous one has all been fired. The time required to consume each portion should be noted, the time being recorded at the instant of firing the first of each new portion. It is desirable that at the same time the amount of water fed into the boiler should be accurately noted and recorded, including the height of the water in the boiler, and the average pressure of steam and temperature of feed during the time. By thus recording the amount of water evaporated by successive portions of coal, the record of the test may be divided into several divisions, if desired at the end of the test, to discover the degree of uniformity of combustion, evaporation and economy at different stages of the test.

PRIMING TESTS.

In all tests in which accuracy of results is important, calorimeter tests should be made of the percentage of moisture in the steam, or of the degree of superheating. At least ten such tests should be made during the trial of the boiler, or so many as to reduce the probable average error to less than one per cent, and the final records of the boiler tests corrected according to the average results of the calorimeter tests. On account of the difficulty of securing accuracy in these tests, the greatest care should be taken in the measurements of weights and temperatures. The thermometers should be accurate to within a tenth of one degree, and the scales on which the water is weighed to within one-hundredth of a pound.

ANALYSES OF GASES.

Measurement of air supply, etc.—In tests for purposes of scientific research, in which the determination of all the variables entering into the test is desired, certain observations should be made which are in general not necessary in tests for commercial purposes. These are the measurements of the air supply, the determination of its contained moisture, the measurement and analysis of the flue gases, the determination of the amount of heat lost by radiation, of the amount of infiltration of air through the setting, the direct determination by calorimeter experiments of the absolute heating value of the fuel, and (by condensation of all the steam made by the boiler) of the total heat imparted to the water.

The analysis of the flue gases is an especially valuable method of determining the relative value of different methods of firing, or of different kinds of furnaces. In making these analyses, great care should be taken to procure average samples

since the composition is apt to vary at different points of the flue, and the analyses should be intrusted only to a thoroughly competent chemist, who is provided with complete and accurate apparatus. As the determination of the other variables mentioned above are not likely to be undertaken except by engineers of high scientific attainments, and as apparatus for making them is likely to be improved in the course of scientific research, it is not deemed advisable to include in this code any specific directions for making them.

RECORD OF THE TEST.

A "log" of the test should be kept on properly prepared blanks, containing headings as follows: —

	Pressures.			Temperatures.					Fuel.		Feed Water.		
Time.	Barometer.	Steam gauge.	Draft gauge.	External air.	Boiler room.	Flue.	Feed water.	Steam.	Time.	Pounds.	Time.	Lbs. or cu. ft.	

REPORTING THE TRIAL.

The final results should be recorded upon a properly prepared blank, and should include as many of the following items as are adapted for the specific object for which the trial is made. The items marked with a * may be omitted for ordinary trials, but are desirable for comparison with similar data from other sources.

Resources of the trials of a . . . . . .
Boiler at . . . .
To determine . . . .

1.	Date of trial	hours.
2.	Duration of trial	hours.

DIMENSIONS AND PROPORTIONS.

3.	Grate-surface wide long area	Sq. ft.
4.	Water-heating surface	Sq. ft.
5.	Superheating surface	Sq. ft.
6.	Ratio of water-heating surface to grate-surface	

AVERAGE PRESSURES.

7.	Steam pressure in boiler, by gauge . .	lbs.
*8.	Absolute steam pressure	lbs.
*9.	Atmospheric pressure, per barometer .	in.
10.	Force of draught in inches of water . .	in.

AVERAGE TEMPERATURES.

*11.	Of external air	deg.
*15.	Of fire-room	deg.
*13.	Of steam	deg.
14.	Of escaping gases	deg.
15.	Of feed-water	deg.

FUEL.

16.	Total amount of coal cousumed . . .	lbs.
17.	Moisture in coal	per cent.
18.	Dry coal consumed	lbs.
19.	Total refuse, dry pounds equals . . .	per cent.
20.	Total combustible (dry weight of coal, item 18, less refuse, item 19) . . .	lbs.
*21.	Dry coal consumed per hour . . .	lbs.
*22.	Combustible consumed per hour . . .	lbs.

RESULTS OF CALORIMETRIC TESTS.

23. Quality of steam, dry steam being taken as unity	
24. Percentage of moisture in steam . . .	per cent.
25. Number of degrees superheated . . .	deg.

WATER.

26. Total weight of water pumped into boiler and apparently evaporated	lbs.
27. Water actually evaporated, corrected for quality of steam	lbs.
28. Equivalent water evaporated into dry steam from and at 212° F.	lbs.
*29. Equivalent total heat derived from fuel in B. T. U.	B. T. U.
*30. Equivalent water evaporated in dry steam from 212° F. per hour . . .	lbs.

ECONOMIC EVAPORATION.

31. Water actually evaporated per pound of dry coal, from actual pressure and temperature	lbs.
32. Equivalent water evaporated per pound of dry coal, from 212° F.	lbs.
33. Equivalent water evaporated per pound of combustible from and at 212° F. .	lbs.

COMMERCIAL EVAPORATION.

34. Equivalent water evaporated per pound of dry coal with one-sixth refuse, at 70 lbs. gauge pressure, from temperature of 100° F., equals item tests 33 X. 0.7249 pounds	lbs.

† Corrected for inequality of water level and of steam pressure at beginning and end of test.

RATE OF COMBUSTION.

35. Dry coal actually burned per sq. foot of grate-surface per hour . . . . . . lbs.

*36.	Consumption of dry coal per hour. Coal assumed with one-sixth refuse.	Per sq. ft. of grate surface . . .	lbs.
*37.		Per sq. ft. of water heating surface .	lbs.
*38.		Per sq. foot of least area for draught.	lbs.

RATE OF EVAPORATION.

39. Water evaporated from and at 212° F. per square foot of heating surface per hour.

*40.	Water evaporated per hour from temperature of 100° F. into steam of 70 lbs. gauge pressure.	Per sq. ft. of grate surface . . .	lbs.
*41.		Per sq. ft. of heating surface . .	lbs.
*42.		Per sq. ft. of least area for draught.	lbs.

COMMERCIAL HORSE POWER.

43. On basis of 30 lbs. of water per hour evaporated from temperature of 100° F. into steam of 70 lbs. gauge pressure ($34\frac{1}{2}$ lbs. from and at 212°) . . . H. P.

44. Horse-power, builders' rating at sq. ft. per horse-power . . . . . .

45. Per cent developed above or below rating per cent.

* NOTE. Items 20, 22, 33, 34, 36, 37, 38 are of little practical value. For if the result proves to be less satisfactury than expected on the actual coal, it is easy for an expert fireman to decrease No. 20 by simply taking out some partly consumed coal in cleaning fires, and thus make a fine showing on that simply ideal or theoretical unit, the "pound combustible." The question at issue is always what can be done with an actual coal, not the "assumed coal" of items 34, 36, 37 and 38.

DEFINITIONS AS APPLIED TO BOILERS AND BOILER MATERIALS.

Cohesion is that quality of the particles of a body which causes them to adhere to each other, and to resist being torn apart.

Curvilinear seams. — The curvilinear seams of a boiler are those around the circumference.

Elasticity is that quality which enables a body to return to its original form after having been distorted, or stretched by some external force.

Internal radius. — The internal radius is one-half of the diameter, less the thickness of the iron. To find the internal radius of a boiler, take one-half of the external diameter and substract the thickness of the iron.

Limit of elasticity. — The extent to which any material may be stretched without receiving a permanent "set."

Longitudinal seams. — The seams which are parallel to the length of a boiler are called the longitudinal seams.

Strength is the resistance which a body opposes to a disintegration or separation of its parts.

Tensile strength is the absolute resistance which a body makes to being torn apart by two forces acting in opposite directions.

Crushing strength is the resistance which a body opposes to being battered or flattened down by any weight placed upon it.

Transverse strength is the resistance to bending or flexure, as it is called.

Torsional strength is the resistance which a body offers to any external force which attempts to twist it round.

Detrusive strength is the resistance which a body offers to being clipped or shorn into two parts by such instruments as shears or scissors.

Resilience or toughness is another form of the quality of

strength; it indicates that a body will manifest a certain degree of flexibility before it can be broken; hence, that body which bends or yields most at the time of fracture is the toughest.

Working strength. — The term "working strength" implies a certain reduction made in the estimate of the strength of materials, so that when the instrument or machine is put to use, it may be capable of resisting a greater strain than it is expected on the average to sustain.

Safe working pressure, or safe load. — The safe working pressure of steam-boilers is generally taken as $\frac{1}{5}$ of the bursting pressure, whatever that may be.

Strain in the direction of the grain, means strain in the direction in which the iron has been rolled; and in the process of manufacturing boiler-plates, the direction in which the fibres of the iron are stretched as it passes between the rolls.

Stress. — By the term "stress" is meant the force which acts directly upon the particles of any material to separate them.

HEAT AND STEAM.

The steam engine is a machine for the conversion of heat into power in motion. The heat is generated by the combustion of fuel; the transmission is accomplished through the agency of steam; the power is made available and brought under control by means of the engine.

The effect of heat upon water is to vaporize it, if there be intensity enough, the heat will, under proper conditions, cause water to boil; the vapor produced by boiling is called steam, and steam under pressure is a product which is the end and aim of that portion of that steam engine known as the boiler and furnace. The steam engine then is to be considered as a form of the heat engine; of which the furnace, boiler, and the engine itself are to be regarded as separate portions of the same mechanism.

The conditions demanded upon economic grounds to secure the highest efficiency in the steam engine are: —

1. A proper construction of the furnace so as to secure the perfect combustion of fuel.

2. The heat generated in the furnace must be transferred to the water in the boiler without loss.

3. The circulation in the boiler must be so complete that the heat from the furnace may be quickly and thoroughly diffused throughout the whole body of water.

4. The construction of an engine that will use the steam without loss of heat, except so much as may be necessary to perform work required of the engine.

5. The recovery of heat from exhaust steam.

6. The absence of friction and back pressure in the working of the engine.

It is superfluous to say that these conditions are not fulfilled in any engine of the present day. At best the combustion of fuel is only approximately perfect, the losses being due to several causes, among which are, — unburned fuel falling through the spaces in the grates and mingling with the ashes. This, with some kinds of coal, and improper firing, amounts to a large percentage of the furnace waste. It is not possible with any present method of setting boilers to transfer all the heat of the furnace to the water in the boiler; nor can there be, for the reason that the temperature of the escaping gases must not be lower than that of the steam in the boilers, or direct loss will result in the radiation of heat from the tubes or flues in the boiler, by thus reheating the gases to the steam temperature. If the steam pressure is 80 lbs. per square inch above the atmosphere, the corresponding temperature due to this pressure is 324° Fahr. The temperature of the escaping gases ought not, therefore, to be less than 350° Fahr., where they leave the boiler flues or tubes to pass off into the chimney. If the temperature of the furnace be taken

at 2,000° Fahr., and the escaping gases at 400° Fahr., it will be seen that one-fifth of the heat generated in the furnace is passing off without performing work. This is a very great loss, and these figures understate, rather than correctly give, the loss from this one source. Efforts have been made to utilize the temperature of these waste gases by making them heat feed-water by means of coils, or by that particular disposition of pipes and connection known as an economizer. Others have turned it into account by making it heat the air supplied the fuel on the grates. Any heat so reclaimed is money saved, provided it does not cost more to get it than it is worth in coal to generate a similar quantity of heat. It is doubtful whether the loss in this particular direction can be brought below 20 per cent of the fuel burned, at least, by any method of saving now known.

The loss by bad firing and by a bad construction of furnace is often a large one. It has been demonstrated experimentally that 20 to 30 per cent of fuel can be saved by a proper construction and operation of the furnace. The direct causes of loss are, too low temperature of furnace for properly burning fuels, especially such as are rich in hydro-carbon gases; or, by the admission of too much cold air over or back of the fire; or, by the admission of too little air under the fire so that carbonic oxide gas is generated instead of carbonic acid gas, the former being a product of incomplete, the latter the product of complete combustion. The relative heating powers of fuel burned, resulting in the production of either of these two gases being as follows: —

	Heat Units.
1 pound of carbon burned to carbonic acid gas . .	14,500
1 pound of carbon burned to carbonic oxide . . .	4,500
Units of heat lost by burning to carbonic oxide .	10,000

It will be seen that here is an enormous source of loss, and all that is required to prevent it is a proper construction of furnace.

Smoke is a nuisance which ought to be prohibited by stringent legislation. There is no good reason for its polluting presence in the atmosphere, defiling everything with which it comes in contact. Smoke regarded as a source of direct loss is greatly overestimated; the fact is, the actual amount of coal lost to produce smoke is very trifling. The presence of smoke indicates a low temperature of furnace or combustion chamber; if the temperature were sufficiently high and the furnace properly constructed, smoke could not be generated. The prevention of smoke is easily accomplished, and with it a more economical combustion of hydro-carbon fuels.

Radiation. — A considerable loss of heat occurs by radiation from the furnace walls; this may be prevented in part by making the walls hollow, with an air space between. If a force blast is used the air may be admitted at the back end of the boiler setting and by passing through between the walls will become heated, and if conveyed into the ash pit at a high temperature will greatly assist combustion and thus tend to a higher economy.

Air required. — In regard to the quantity of air required, it will vary somewhat with the fuel used, but in general, 12 pounds of air are sufficient to completely burn one pound of coal; practically, however, 15 to 25 pounds are furnished, being largely in excess of that which the fire can use, and must pass off with the gases as a waste product. This surplus air enters cold and leaves the furnace heated to the same temperature as that of the legitimate and proper products of combustion, and thus directly operates to the lowering of the furnace temperature.

Measurement of heat. — A heat unit is that quantity of heat necessary to raise the temperature of one pound of water one degree, from 39° to 40° Fahr., this being the temperature of the greatest density of water. A thermal unit, a heat unit, or unit of heat, all mean the same thing. Experiments have been made to determine the mechanical equivalent of a heat unit, and it is

found to be equal to 778 pounds raised one foot high. This is sometimes called "Joule's equivalent," after Dr. Joule, of England; it is also known as the dynamic value of a heat unit.

Knowing the number of heat units in a pound of coal enables us to calculate the amount of work it should perform. Let us suppose a pound of coal to be burned to carbonic acid gas, and to develop during its combustion 14,000 heat units, then: 14,000 x 778 equals 10,892,000 foot pounds.

That is to say, if one pound of coal were burned under the above conditions it would have a capacity for doing work represented by the lifting of ten millions of pounds one foot high against the action of gravity. Suppose this to be done in one hour, then we should expect to get from one pound of coal an equivalent of 5.45 H. P. It is well known that only a very small fraction of such equivalent is secured in the very best modern practice. The question is, where does this heat go, and why is it so small a portion of it is actually utilized? The losses may be accounted for in several ways, and, perhaps, as follows: —

The heat wasted in the chimney	25 per cent.
Through bad firing	10 "
Heat accounted for by the engine (not indicated)	10 "
Heat by exhaust steam	55 "
	100 per cent.

This is about 2 pounds of coal per hour per indicated horse power, which is regarded as a very high attainment, and is seldom reached in ordinary cut-off engines. It requires good coal, good firing, and an economical engine to get an indicated horse power from two pounds of coal burned per hour. As coal varies in quality it is a better plan to deduct the ashes and other incombustible matter, and take the net combustible as a basis of comparison. The best coal when properly burned

is capable of evaporating 12 pounds of water from and at a temperature of 212° Fahr. The common evaporation is about half that amount, and with the best improved furnaces and careful management, it is seldom that 10 pounds of water is exceeded, and is to be regarded as a high rate of evaporation. In experimental tests, 12 pounds have been reported, but it is doubtful whether there is any steam boiler and furnace which is constantly yielding any such results.

Circulation of water in a boiler is a very important feature to secure the highest evaporative results. Other things being equal, the boiler which affords the best circulation of water will be found to be the most economical in service. Circulation is greatly hindered in some boilers by having too many tubes; in others, by introducing in the water space of the boiler too many stays and making the water spaces too narrow. To secure the highest economy there must be thorough circulation from below upwards, in the boiler. There is no doubt that a great deal of heat is lost because the construction is such as to hinder a free flow of water around the tubes and sides of the boiler.

The construction of an engine that will use steam without loss of heat, except so much as may be necessary to perform work required of it, is a physical impossibility. Among the sources of loss in an engine are: radiation, condensation of steam in unjacketed cylinders, and the enormous loss of heat occasioned by exhausting the steam into the atmosphere.

Radiation is usually classed among the minor losses in a steam engine. There is a considerable loss of heat caused by radiation from steam boilers and pipes exposed to the atmosphere, and not protected by a suitable covering. Much of this heat may be saved by employing a non-conducting material as a covering, which, though not preventing all radiation, will save enough heat to make its application economical. It is well known that some bodies conduct and radiate heat less rapidly than others, but it

must not be understood that the absolute value of such a covering is inversely proportioned to the conducting power of the material employed, because, in its application, the outer surface is enlarged and the radiation will be going on less actively at any given point, but the enlarged surface exposed reduces somewhat the apparent gain.

SELECTION OF A BOILER.

The selection of a boiler for a particular service will naturally suggest the following questions: —

1. What kind of a boiler shall it be?
2. Of what material shall it be made?
3. What size shall it be in order to furnish a certain power?

In reply to the first question, it is to be expected there will be wide differences of opinion, varying with the locality, usage, and service for which it is intended. One of the first things to be taken into account in the selection of a boiler is the quality of water to be used in it for generating steam. If the water is pure, then it makes little difference what kind of boiler be selected, so far as incrustation affects selection. If the water is hard and will deposit scale upon evaporation, then a boiler should be selected which will admit of thorough inspection and removal of any deposit formed within it.

For hard water, the ordinary flue boiler will be found a good one, as it is favorable to a thorough circulation of water, and permits easy access to all parts of it for examination and cleaning. It does not, however, present the extent of heating surface for a given space that tubular boilers offer; but with hard water the boiler is quite as economical if kept in good condition.

The difficulty with tubular boilers when used in connection with hard water is that the tubes will in a short time become coated with scale; this prevents the transmission of heat, not only, but impairs the circulation of the water around them.

Both of these are opposed to economy in the fact that it requires more coal to generate a given weight of steam in the first case; and second, by reason of deficient circulation the plates over the fire are likely to become overheated and burnt and so become dangerous; thus directly contributing to accident or disaster.

The matter of circulation in boilers is one which should have careful attention in making a selection. There is little trouble in this regard with any of the ordinary types of boilers so long as they are clean and new, and properly proportioned. Nor is there likely to be any difficulty thereafter if the water is soft and clean. Circulation is often seriously impaired by putting in too many tubes in a boiler, the effect of which is to so fill up the space that the heated particles of water forcing their way upwards from below meet with so much resistance that they can hardly overcome it, and the result is that a boiler does not furnish from one-fourth to one-half as much steam for a given weight of fuel as it should, from this very cause.

Boilers intended for use in distant localities where the facilities for repairs are meager or entirely wanting, and fuel low priced, should be of the simplest description. Cylinder boilers or two-flue boilers will perhaps be found most suitable. These are largely used by coal miners, blast furnaces, saw mills, and other branches of industry, which must, of necessity, be removed from the larger towns and engineering work shops.

In selecting a boiler for a mill of any kind where they burn shavings or offal, or any other place in which the fuel is of a similar description and the firing irregular, there should be large water capacity in the boiler that it may act as a reservoir of power in much the same way that a fly wheel acts as a regulator for a steam engine. It is a common notion among wood workers that firing with shavings or light fuel is "easy on the boiler." There is abundant reason to doubt this. The suddenness and rapidity with which an intense fire is kin-

dled in the furnace, filling all the furnace space and the tubes with flame, and with an intense heat which envelops all within the limits of draft opening, continuing thus for a few minutes only, and as suddenly going out, can hardly be regarded as the ideal furnace. Yet there are thousands of just such furnaces at work, and it is altogether probable that little or no change will be made in them by this class of manufacturers, at least in the near future. In regard to the selection of a boiler for this service, we are brought back again to the question of hard or soft water. The decision should be largely influenced by this, but whatever type of a boiler is selected there should be a surplus of boiler power of at least 20 per cent, that is, if a 50 horse-power boiler is needed to do the work, put in one of 60 horse-power; this will prevent the fluctuations of speed in the engine which are sure to follow a reduction of boiler pressure.

This increase in boiler power ought not to be simply that of tube surface, but should also include extra water space. The reserve power of a boiler is in the water heated up to a temperature corresponding to the steam pressure; when this pressure is lowered, the water then gives off steam corresponding to the lower pressure; the more water the more steam; and in this way the water in the boiler stores up heat when overfired, to give it off again when the fire is low, and so acts a regulator of pressure, a thing that extra tube surface cannot do. This kind of firing is apt to induce priming, and for this reason a boiler should be selected having a large water surface. Horizontal boilers are, in general, to be preferred over vertical ones for mills, because of the larger water surface exposed in proportion to the heating surface. If a tubular boiler is selected, the water line above the tubes should be not higher than two-thirds the diameter of the boiler measured from the bottom, and the boiler should be made having the upper edge of the top row of tubes at least three inches below this; there should also be a clear space up through the center of

the boiler of sufficient width to insure a perfect circulation of water.

Horizontal tubular boilers are to be recommended when pure soft water is used. They combine at once the qualities of great strength without excessive bracing, large heating surface, high evaporative capacity without liability to priming, and are convenient of access for external and internal examination when set in the furnace.

Firebox boilers, or locomotive boilers, as they are commonly called, are best adapted for small powers and with a fuel which deposits but little soot in the tubes. This kind of boiler is supplied with portable or agricultural engines and is very well adapted for that particular service. In canvassing the desirability of this kind of a boiler for stationary use, we must again refer to the kind of water to be used in it. If the water is soft and clean there is then no particular objection to a boiler of this construction being used for small powers; if the water is hard and will form scale, it ought not to be chosen, but a flue boiler selected instead.

Vertical boilers are used in great numbers for small engines, heating, etc. They have the merit of being compact and low priced. A common defect in the construction of this kind of boiler is that too many tubes are put in the head in the fire box, thereby preventing a proper circulation of water between them. This defect in construction induces priming, with all its attendant annoyances and dangers. This style of boiler is not suited to hard water, but pure soft water only. These boilers should be provided with hand holes above the crown sheet and around the bottom of the water legs; at least three at each place mentioned. In regard to the material of which a boiler shall be made there is but the simple choice between iron and steel.

Steel for boilers should not be of too high tensile strength; 55,000 to 60,000 pounds tensile strength per square inch makes

the best boilers. If the steel is of too high a grade it will take a temper, and, therefore, is utterly unfit for use in steam boilers; if the steel is of too low tensile strength it is apt to be loose or spongy. Among the advantages steel possesses over iron may be mentioned the circumstance that it is a practically homogeneous material when properly made and rolled, consequently, it is nearly as strong in one direction as it is in another. In this respect, steel is superior to iron plate of equal thickness, because the latter is made up of several pieces of iron welded together and in rolling into the plate it becomes fibrous, and thus of unequal strength, being greatest in the direction of the fiber, and least, when tested across it.

BOILER TRIMMINGS.

The common trimmings to a steam boiler are a safety valve, feed and blow-off pipe, steam pipe, gauge cocks, glass water gauge and steam gauge; to which may be added a steam drum or dome and a mud drum. There are numerous other devices, which are attached to boilers such as safety gauges, alarms, fusible plugs, automatic dampers, etc.; many of these are very serviceable and are well liked by those using them.

Safety valves should always be large enough to permit the escape of all the steam a boiler is capable of making and each boiler should have its own safety valve rather than connecting two or more boilers together and depending on one valve for the whole. The valve and seat should be made of hard gun metal, or any other composition that will not rust and stick fast. At one time it was quite a common thing to see a brass valve fitted to a cast-iron seat; this is wrong, for the rusting of the iron would fix the valve so tightly that the boiler would be in constant danger of rupture from over pressure. For stationary boilers the common ball and lever safety valves are generally used. For stationary boilers it is immaterial whether the safety valve be fitted with

lever and weight, or whether it be fitted with a spring. The former is the usual manner of loading a safety valve and has but few objections. For portable engines and locomotives safety valves are loaded with springs, which by suitable adjustment may be made to blow off at any desired pressure.

The following rule is that enforced by the U. S. Government in fixing the area of safety valves for ocean and river service, when the ordinary lever and weight safety valve is employed: —

Rule. — When the common safety valve is employed it shall have an area of not less than one square inch for each two square feet of grate surface.

Another rule is to multiply the pounds of coal burned per hour by 4; this product is to be divided by the steam pressure, to which a constant number 10 is added.

EXAMPLE: What would be the proper area for a safety valve for a boiler having a grate surface 5 feet square and burning 12 pounds of coal per hour per square foot of grate; the steam pressure being 75 pounds per square inch?

5 x 5 equal 25 square feet of grate.

25 x 12 equal 300 lbs. of coal per hour.

300 x 4 equal 1200.

75 plus 10 equal 85 equal steam pressure with 10 added, then 1200 ÷ 85 equal 14.11 inches area, or $4\frac{1}{4}$ inches diameter.

A feed pipe should be at least twice the area over that which is regarded as simply necessary to supply the boiler with water, as sediment or scale is likely to form in it, which will materially reduce its area. In localities where the water is hard the feed pipes should be disconnected near the boiler and examined occasionally to ascertain whether or not scale is forming in them.

In general, the sizes of feed pipes leading from the pump to the boiler are fixed by the size of tap used by the maker of the pump. It is not well to reduce the diameter of the pipe and the size should be the same throughout. Care should be exer-

cised in putting pipes in place that no strain be brought upon them by imperfect fitting, as it is certain to lead to leaky joints at some time or other. It is also desirable that the pipes be as short and straight as possible. Feed pipes should never be placed under ground if it is possible to make any different disposition of them. In locating pipes it is desirable to arrange for the expansion of the boiler, as well as for that of the pipes themselves. In selecting a pump it should have a much larger capacity than that needed to supply the boiler, as there are many things which affect the working of a pump, such as a defective suction pipe, leaky valves, etc. It is the practice of most manufacturers to give the capacity of their pumps in gallons of water delivered per minute, from which it is easy to select a suitable size; but the speed given in the tables at which the pump is to run is generally faster than that which it is desirable to run them. As a general thing, and without referring to any particular maker or design, it is a good plan to select a pump having four times the capacity actually needed for the boiler; then the speed may be reduced to half that given in the table, and will require less repairs, and will be a more satisfactory purchase in the long run.

In selecting an injector or inspirator, the size should not greatly exceed that actually required to supply the boiler. In making the steam connections the pipes should start from the steam space of the boiler and should not be branches merely from the other steam pipes; neither should the diameters of the pipes be less than that which the instrument calls for. The pipes should be as short and straight as practicable; abrupt bends should always be avoided in the suction pipes. If the water is taken from a place in which there are floating particles of wood, leaves, etc., a strainer should be used; a large sheet metal box with perforated sides, makes a good strainer; the openings ought not too greatly exceed an eighth of an inch in diameter, and should be several times the area of the suction pipe.

A check valve should be fitted with a valve between it and the boiler, so that in the event of its not working satisfactorily it may be taken apart, cleaned and replaced without stopping for examination or repairs.

The blow-off pipe should be so arranged that it will entirely drain the boiler of water; it is also a good plan to set a boiler with a slight inclination toward the blow-off pipe that it may be thoroughly drained; an inclination of two inches in twenty feet works well in practice. The blow-off pipe is usually fitted at the back end of the boiler.

The steam pipe may be connected at any convenient point on the top of the boiler. If the boiler is to furnish steam for an engine only, the common practice is to make the diameter of the pipe one-fourth that of the cylinder. The steam pipe should be as short and straight as possible. If bends are to be introduced in steam pipes it is better to have a long curved bend than the abrupt right-angle fitting usually employed for the purpose. It is also a good plan to provide a stop-valve next to the boiler to shut off the steam and prevent it condensing in the steam pipe at night, or other long stoppages.

The gauge cocks should not be less than three in number, and may be of any of the various kinds now in the market. For stationary boilers, the Mississippi gauge cock is, perhaps, as good as any. For portable engines a compression gauge-cock is, perhaps, the best. The lower gauge-cock should be at least 2″ above the tubes or crown sheet, the middle 2″ above the first ordinary water line, the upper 2″ above the 2 on 2″ to 3″, depending on the size of the boiler.

A glass water gauge should be provided for each boiler and should be so located that the water level in the boiler when at the lower end of glass shall be one inch above the top of flue. When glass gauges are so fitted the fireman can always tell at a glance just how much water he has above the flues or crown sheet; it

also permits the easy test of accuracy by trying the gauge-cocks with the water at a certain known level. Too much dependence must not be placed on the glass water-gauge alone, but should be used in connection with the gauge-cocks.

A steam gauge is a very important appendage to a steam boiler, and should be chosen with special reference to accuracy and durability. The ordinary gauges now in the market are the bent tube and the diaphragm gauges. It matters little which of the two kinds is selected, provided it is a good and first-class gauge. A steam gauge should be compared with a standard test gauge at least once a year, to see that it is correct. The importance of this will be fully apparent when it is known that it furnishes the only means by which the fireman is to judge of the steam pressure in the boiler. A siphon should be attached to every gauge, and provision should also be made for draining the gauge or siphon, to prevent freezing when steam is off the boiler. Neglect of this may endanger the accurate reading of the steam gauge and render it useless.

Steam dome. — This is a reservoir for steam riveted to the upper portion of the shell and communicated by a central opening with the steam space in the boiler. When this reservoir forms a separate fixture and is attached to the boiler by cast or wrought iron nozzles, it is then called a *steam drum*. The latter answers all the purposes for stationary boilers that the former does, and is to be preferred because of the smaller openings in the shell of the boiler. A considerable number of boiler explosions have been traced directly to the weakness of the shell, caused by the large opening in and imperfect staying of the shell underneath the dome. When a dome is employed and has a large hole underneath, the strength of the shell is impaired in two ways: 1. By reducing the longitudinal sectional area of shell through the center of opening cut for it, which weakness cannot wholly be made good by a strengthening ring around the opening. 2. By causing

a tension equal to that on the crown area of steam dome, upon the annular part of the shell covered by the flange of the dome. The weakest part of the boiler shell will be where the distance from rivet hole at the base of the dome to edge of plate is least. It is difficult, owing to the complex nature of the strains, to form a rule whereby to determine how much the strength of the shell is impaired by using a dome; but it is quite apparent from general experience that they are in many cases a source of weakness, and the larger the dome connection with the shell, the greater the liability to rupture. This tendency to rupture is due to the fact that the dome, with its connecting flange, is practically inelastic; that portion of the shell of the boiler covered by the dome is, as soon as the pressure is introduced on both sides of the plate, simply a curved brace. The pressure of the steam in the boiler has a tendency to straighten the shell under the dome and thus brings about a series of complex strains, which are not easily remedied by any system of bracing, so that on the whole it is preferable to use a small connecting nozzle with a drum above it, rather than to rivet a large dome directly to the shell.

Dry pipe. — This is a pipe having numerous small perforations on its upper side, and inserted in the upper part of the steam space of the boiler. This pipe does not dry the steam, but acts mechanically by separating the steam from the water when the latter is in a violent state of agitation, and is liable to be carried in bulk toward or into the steam pipe. The object of these numerous small holes in the pipe is that a small quantity of steam may be taken from a large number of openings at one time, and thus carried over a larger extent of surface than that afforded by a single opening, and by this single device checking the tendency to priming.

Steam boiler furnaces are receiving more attention now than perhaps ever before. The question of economy of fuel is being closely studied, and there is now an effort to save much of the

heat, which had formerly been allowed to go to waste. The main thing in furnace construction is to get perfect combustion. Without this there must be of necessity a great loss constantly going on. There are some losses which it is difficult to prevent, for example — the loss by the admission of too much air in the ash pit; the loss by incomplete combustion; the loss occasioned by the heated gases escaping up the chimney; the loss by radiation; but, chief among these, is that of incomplete combustion. To burn a pound of coal requires about twenty-four pounds of air, or, say 300 cubic feet. Most boiler settings permit from 200 to 300 feet to pass through the fire. It is needless to point out the great source of loss arising from this one cause alone. This may be prevented in a measure by having a suitable damper in the chimney, and regulating the flow of escaping gases by it, instead of the ash pit doors. If the furnace is so constructed that the fuel is imperfectly burned, so that carbonic oxide instead of carbonic acid gas is formed, the loss is very great. This results often from too little air supply and too low temperature in the furnace. The furnace doors should be provided with an opening leading into the space between the door proper and the liner; this opening ought to have a sliding or revolving register by which the admission of air may be controlled. By this means, the quantity of air admitted above the fire may be adjusted to its needs by a little attention on the part of the fireman. The liner to the furnace door should have a number of small holes in it, rather than a solid plate, with a space around the edges. Great care should be exercised in the construction of furnace walls, that the materials and workmanship be good throughout. The entire structure should be brick. The outer walls may be of good hard red brick, but the interior walls, around the furnace and bridge wall, should be of fire brick. The best quality of fire brick for withstanding an intense heat are very, very strong and tenacious; the structure is open and they are free from black

spots, due to sulphuret of iron in the clay; if well burned they will not be very light colored on the outside, and will have a clear ring when struck.

Fire brick should be dipped in a thin mortar made of fire clay, rather than in a lime and sand mortar, such as is used in ordinary red brickwork. In laying up these portions of a boiler furnace requiring fire brick, provision should be made in the original wall for replacing the fire brick and without disturbing the outer brickwork.

CARE AND MANAGEMENT OF A BOILER.

It is not enough that a boiler be of approved design, made of the best materials, and put together in the best manner; that it have the best furnace and the most approved feed and safety apparatus. These are all desirable, and are to be commended, but cleanliness and careful management are quite essential to getting high results, and are also conducive to long use in service.

Pumps. — Special attention should be given at all times to the feed and safety apparatus; the pumps should be in good working order; it is preferable that they be independent steam pumps rather than pumps driven by the engine, or by a belt; they should be kept well packed and the valves in good condition.

Firing. — Kindle a fire and raise steam slowly; never force a fire so long as the water in the boiler is below the boiling point. The fire should be of an even height, and of such a thickness as will be found best for the particular fuel to be burned, but should be no thicker than actually necessary. In regard to the size of coal used, that will depend upon circumstances. If anthracite coal is used, it should not, for stationary boilers, be larger than ordinary stove coal. For bituminous coal, which is always shipped in lumps as large as can be conveniently handled, the size will vary somewhat in breaking, but it may in general be used in larger lumps than anthracite. If the coal is likely to cake in burn-

ing, the fire should be broken up quite frequently with a slice bar, or it will fuse into a large mass in the center of the furnace and lower the rate of combustion. If the coal is likely to form a considerable quantity of clinker, or enough to become troublesome, it may be advantageous to increase the grate area and thus lower the rate of combustion per square foot of grate, and have a fire of less intensity. The fire should be kept free from ashes, and the ash pit should be kept clean. Whenever the fire door of a steam boiler furnace is opened, the damper should be closed to prevent the sudden reduction of temperature underneath, which is likely to injure the boiler by contraction, and thus render it likely to spring a leak around the riveted joints. Some firemen are very careless in this respect, and there is little doubt that many a disagreeable job of repairing a leaky seam might be prevented by this simple precaution.

Gauge cocks should be used constantly to keep them free from any accumulation of sediment. It is a very common practice to rely wholly on the indications of the glass water gauge for the water level in the boiler. This is all wrong and should be discontinued, if once begun. The glass water gauge serves a very useful purpose, but it should not be wholly relied on in practice. In using the ordinary gauge cocks, the ear more than the eye, detects the water level, and thus acts as a check on the indications given by the glass gauge.

Water gauges should be tested several times during the day to see that they are clear, and to keep them free from any sediment likely to form around the lower opening to the water in the boiler. If this is not attended to, the water gauge is likely to indicate a wrong water level and a serious accident may be the result.

Steam or pressure gauges are likely to become set after long use and should be tested at least once, or better still, twice a year by a standard gauge known to be correct. They should also be

tested every few days if the boilers are constantly under steam by turning off the steam and allowing the pointer to run back to zero. If there are two or more boilers set together in one battery, and each boiler has its own steam gauge, and which will, starting from the zero point, indicate the same pressure on all the gauges, they may be assumed to be correct.

Blow-off cocks or valves should be examined frequently and should never be allowed to leak. In general a cock is to be preferred to a valve, but both is safer than one; if the latter is selected it should be some one of the various "straight-way valves," of which there are now several in the market. If the cock is a large one, and especially if it has either a cast iron shell or plug, it should be taken apart after each cleaning out of the boilers, examined, greased with tallow and returned.

Blowing out. — This should be done at least once a day, except in the very rare instances in which water is used that will not form a scale. The water should not be let out of a boiler or boilers until the furnace is quite cold, as the heat retained in the walls is likely to injure an empty boiler directly by overheating the plates, and indirectly by hardening the scale within the boiler. Bad effects are likely to follow when a boiler is emptied of its water before the side walls have become cool; but greater injury is likely to result when cold water is pumped into an empty boiler heated in this manner. The unequal contraction of the boiler is likely to produce leaky seams in the shell and to loosen the tubes and stays. It is a better plan to allow the boiler to remain empty until it is quite cold, or sufficiently reduced in temperature to permit its being filled without injury. Many boilers of good material and workmanship have been ruined by the neglect of this simple precaution.

Fusible plugs should be carefully examined every six months, as scale is likely to form over the portion projecting into the water space. It is only a question of time when this scale

would form over the end of the plug, and thick enough to withstand the pressure of steam and thus fail in the accomplishment of the very object for which it was introduced. This applies especially to the fusible plugs inserted in the crown sheets of portable engine boilers.

Cleaning tubes. — This should be done every day if bituminous coal is used. A portable steam jet will be found an extremely useful contrivance which will keep them reasonably clean by blowing out the loose soot and ashes deposited in the tubes. Every two or three days, or at least once a week, a tube scraper or stiff brush should be used to take out all the ashes or soot adhering to the tubes and which cannot be blown out with the jet. Flues may be cleaned the same way but will not require to be done so frequently.

Low water. — If from any cause the water gets low in the boiler, bank the fire with ashes or with fresh coal as quickly as possible, shut the damper and ash pit doors and leave the fire doors wide open; do not disturb the running of the engine but allow it to use all the steam the boiler is making; do not under any circumstances attempt to force water in the boiler. After the steam is all used and the boiler cooled sufficiently to be safe, then the water may be admitted and brought up to the regular working height; the damper opened and the fires allowed to burn and steam raised as usual; provided, no injury has been done the boiler by overheating.

Foaming and priming are always troublesome and often dangerous. Some boilers prime almost constantly, because of their bad proportion, and will require the constant care of the person in charge, especially at such times as the engine may be using the steam up to the full capacity of the boiler. In a case of this kind, an increase in pressure will often check, but will not entirely prevent it; nothing short of an increase of water surface, or a better circulation of water, or a larger steam room will afford a

complete remedy. If the foaming or priming is due to a sudden liberation of steam, or on account of impure feed water it may be checked by closing the throttle valve to the engine and opening the fire door for a few minutes. The surface blow may be used with advantage at this time, by blowing off the impurities collected on the surface of the water. The feed pump may be used if necessary, but care should be exercised that too much cold water be not forced into the boiler, and thus lose time by having to wait for the accumulation of the regular steam pressure required for the engine. The dangers attending foaming or priming are: the laying bare of heating surfaces in the boiler, and of breaking down the engine by working water into the cylinder. The commonest damage to the engine being either the breaking of a cylinder head, or the cross-head, or the breaking of the piston. When boilers are new and set to work for the first time priming is a very frequent occurrence; in fact, it may be said that for the first few days there is always more or less of it. All that is needed during this time is a little care on the part of the attendant to see that the water is kept up to the required level in the boiler; it is also recommended that the throttle valve to the engine be partially closed to prevent any very great variation of pressure in the boiler, and thus prevent water passing over with the steam in such quantities as to become dangerous. If a boiler continues to prime after it has had a week's work and then thoroughly cleaned, the causes are to be attributed to other than the grease and dirt in it, which are inseparable from the manufacture. As already said, priming may be caused by a sudden reduction of pressure; that is, a boiler may be working smoothly and well with, say, 80 pounds pressure; if an increase of load be suddenly applied to an engine so as to reduce the pressure to 70 or 60 pounds, this sudden reduction of pressure will almost always cause priming; the less the steam space in the boiler, the greater the tendency to prime, and the greater the

difficulty in checking it. The only permanent cure for this is more boiler power; as a temporary expedient, the engine should be throttled sufficiently to make the drain upon the boiler constant instead of intermittent. If the duty required of an engine is irregular, the steam pressure should be carried higher; in any case similar to the above, it is recommended that the pressure be increased to 90 or 100 pounds and the throttling to begin with the increased drain upon the boiler. But this is at best a mere makeshift, and a larger boiler power becomes imperative both on the score of economy and safety.

WATER FOR USE IN BOILERS.

Water is never pure, except when made so in a laboratory or by distillation; the impurities may be divided into four classes: 1. Mechanical impurities. 2. Gaseous impurities. 3. Dissolved mineral impurities. 4. Organic impurities.

(a) Mechanical impurities may be both mineral and organic. The commonest suspended impurity in water is mud or sand; these may be removed by filtration or by allowing the water to stand long enough to let them settle to the bottom of the tank or cistern and then carefully drawing the water from the top, and without disturbing the bottom.

(b) Gaseous impurities in water vary somewhat according to the localities from which they are obtained. The commonest gases found in the water are an excess of oxygen, nitrogen and carbonic acid. These have no effect on water intended for steam boilers.

(c) Dissolved mineral impurities in water are of the most varied description, and are almost always found in it. Among these are found salts of iron, sulphate and carbonates of lime; sulphate and carbonates of magnesia; salt and alkalies, such as soda, potash, etc.; acids, such as sulphuric, phosphoric, and others. All of these are more or less injurious to steam boilers. The most objectionable are the salts of lime and magnesia, which impart to water that property known as hardness. When such

water is used in a steam boiler a scale will gradually form, which will, in a short time, become very troublesome.

(d) Organic impurities are present, to a certain extent, in most waters. They are sometimes present in the water in sufficient quantities to give it a very decided color and taste.

The presence of organic matter in water is often dangerous to health, and may be a means of spreading contagious diseases, but has little or no bad effect in any water used for steam boilers. In general, water is regarded by engineers as being either soft, hard or salt.

Ebullition is the motion produced in a liquid by its rapid conversion into vapor. When heat is applied to the bottom of a boiler, the particles of water in contact with the plates become heated and immediately expand, and becoming specifically lighter, pass upwards through the colder body of water above ; the heat of the furnace is in this way diffused throughout the whole body of water in the boiler by a translation of the particles of water from below upwards, and from top to bottom in regular succession. After a time this liquid mass becomes heated to a degree in which there is a violent agitation of the whole body of water, steam is given off and it is said to boil. The temperature at the boiling point of water, at ordinary atmospheric pressure, is 212° Fahr., and increases as the pressure of steam above it increases.

Distilled water for boilers is not to be recommended without some reservation. Chemically pure water, and especially water which has been redistilled several times, has a corrosive action on iron which is often very troublesome. The effect on steel plates by the use of water several times redistilled, such, for example, as that supplied for heating buildings, is well known ; information is yet wanting which shall point with certainty to the exact change which the water undergoes and explain why its action on or affinity for steel is so greatly intensified. It has been suggested as a means of neutralizing this corrosive action of the water, to

introduce with the feed other water, which shall have the property of forming a scale and continuing it long enough and at such intervals as will permit the formation of a thin scale in the interior of the boiler. However objectionable this may seem at first sight, it is at present the best practical solution of the difficulty.

Scale is a bad conductor of heat and is opposed to economical evaporation. It is estimated that a thickness of half an inch of hard scale firmly attached to a boiler plate will require a temperature of about 700° Fahr. in the boiler plate in order to raise and maintain an ordinary steam pressure of 75 pounds. The mischievous effects of accumulated scale in the boiler, especially in the plates immediately over the fire, are: (1) preventing the water from coming in contact with the plates, and thus directly contributing to the overheating of the latter; and (2) by causing a change of structure in the plates and the consequent weakening brought about by this continual overheating, which would, in a short time, render an iron or a steel plate wholly unfit for use in a steam boiler. The two principal ingredients in boiler scale are lime and magnesia. The lime, when in combination with carbonic acid, forms carbonate of lime; when in combination with sulphuric acid, it then becomes sulphate of lime. This is also true of magnesia.

Carbonate of lime will form in the boiler as a loose powder, which is held mechanically in suspenion; while in this stage it may be blown out of the boiler without injury to it; but it is seldom that a pure carbonate is formed in the boiler as there are other impurities in the water with which it combines to form a hard scale. This is especially true in such waters as also contain sulphate of lime in solution. This fine powder (carbonate of lime), will form a hard scale should any adhere to the sides or bottom of a boiler; in any case where the boiler is blown out dry while the furnace walls are still hot; and this, in itself, forms an excellent reason why boilers should stand until the furnace walls

are cold before blowing out. When emptied, nearly or all of this slushy deposit may be washed out of the boiler by means of a hose.

Sulphate of lime is not so easily got rid of, as it is heavier than carbonate of lime and adheres to the plates while the boiler is at work. It is the most troublesome scale steam engineers have to deal with; it is very difficult to remove and by successive layers becomes dangerous, on account of the thickness to which it eventually accumulates.

The carbonates of lime and magnesia may be largely arrested by passing the feed water through a suitable heater and lime extractor. It must be apparent to every one that any device which will accomplish this is a very desirable attachment to a steam boiler. As it is not possible to eliminate all the foreign matter in the water from it, recourse is often had to the use of solvents and chemical agencies for the prevention of scale. Some of these are very simple and within easy reach; others are surrounded by an atmosphere of uncertainty and the real nature of the compound is hidden under a meaningless trade-mark. For carbonate of lime, potatoe has been found to be very serviceable in preventing the formation of scale; its action appears to be that of surrounding the particles of lime with a coating of starch and gelatine, and thus preventing the cohesion of these particles to form a mass. Various astringents have been used for this purpose, such as extracts of oak and hemlock bark, nutgalls, catechu, etc., with varying success.

Carbonate of soda has been used and with very great success in some localities, not only in preventing, but in actually removing scale already formed. It acts on carbonate of lime, not only, but on the sulphate also. It is clean, free from grit, and is quite unobjectionable in the boiler; one or more pounds per day, depending on the size of the boiler, may be admitted through the pump with the feed water; or admitted in the morning before

firing up, by simply mixing with water and pouring into the boiler through the safety valve or other opening.

Tannate of soda has been similarly employed and is an excellent scale preventive. It will also act as a solvent for scale already formed in the boiler, acting on sulphate as well as carbonate of lime.

Crude petroleum has been found very beneficial in removing the hard scale composed principally of sulphate of lime.

Zinc in steam boilers. — The employment of zinc in steam boilers, like that of soda, has been adopted for two distinct objects: (1) to prevent corrosion, and (2) to prevent and remove incrustation. To attain the first object, it has been used chiefly in marine boilers, and for the second, chiefly in boilers fed with fresh water. In order that the application of zinc in marine boilers may be effective, it is necessary that the metallic contact should be insured. If galvanic action alone is relied upon for the protection of the plates and tubes, it will doubtless be diminished materially by the coating of oxide that exists between all joints of plates, whether lapped or butted, and also between the rivets and the plates. Assuming the preservative action of zinc to be proved when properly applied, we have now two systems for preventing the internal decay of marine boilers, viz.: allowing the plates and tubes to become coated with scale, and employing zinc. It remains to decide which of these two systems is the best with respect to economy and practicability.

We come now to consider the use of zinc for preventing and removing incrustation.

At one time it was considered that the action of zinc in preventing incrustation was physical or mechanical. The particles of zinc, as it wasted away, were supposed to become mixed amongst the solid matter precipitated from the water, in such a manner as to prevent it adhering together, so as to form a hard scale; or the particles of zinc were supposed to become deposited

upon the plates, and so prevent the scale from adhering to them. Then it was suggested that the zinc acted chemically, and now, it is the generally received opinion that its action is galvanic in preventing incrustation as well as in preventing corrosion. When the water contains an excess of sulphates or chlorides over the carbonates, the acid of the former will form soluble salts with the oxide of zinc, the surface of the zinc will be kept clean, and the galvanic current, to which the efficiency of the zinc is due, will be maintained. On the other hand, should there be a preponderating amount of carbonates, the zinc will be covered first with oxide, then with carbonates and its useful action arrested and stopped. It is quite as important that the zinc should be in metallic contact with the plates when used to prevent incrustation, as when employed to prevent corrosion. The application of zinc for the former purpose should never be attempted without first having the water analyzed in order to ascertain whether it is likely to be effective. The use of zinc in externally fired boilers should be attempted with great caution, as when efficacious in preventing the formation of a hard scale, it is liable to produce a heavy sludge that may settle over the furnace plates and lead to overheating. On the whole we cannot but regard the evidence as to the effect of zinc upon incrustation as being very conflicting.

Leaks should be stopped as soon as possible after their discovery; the kind of leak will indicate the treatment necessary. If it occurs around the ends of the tubes, it may be stopped by expanding the tubes anew; if in a riveted joint, it should be carefully examined, especially along the line of the rivets and care should be exercised in determining whether there is a crack extending from rivet to rivet along the line of the holes; should this prove to be the case, the boiler is then in an extremely dangerous condition and under no circumstances should it be again fired up until suitable repairs have been made which will insure its safety.

Blisters occur in plates which are made up of several thicknesses of iron and which from some cause were not thoroughly welded before the final rolling into plates. When such a plate comes in contact with the heat of the furnace the thinnest portion of the defective plate "buckles" and forms what is called a blister. As soon as discovered, there should be thorough examination of the plate and if repairs are needed there should be as little delay as possible in making them. If the blister be very thin and altogether on the surface it may be chipped and dressed around the edges; if the thickness is equal or exceeds $\frac{1}{16}''$ the blister should be cut off and patched, or a new plate put in.

Patching boilers. — When a boiler requires patching it is better to cut out the defective sheets and rivet in a new one; or if this cannot be done, a new piece large enough to cover the defect in the old sheet may be riveted over the hole from which the defective portion has been cut. If this occurs in any portion of the boiler subject to the action of fire, the lap should be the same as the edges of the boiler seams, and should be carefully calked around the edges after the riveting. Whenever the blisters occur in a plate, patching is a comparatively simple thing as against the repairs of a plate worn by corrosion. In the latter case, the defective portions of the plate should be entirely removed and the openings should show sound metal all around and of full thickness. If this cannot be obtained within a reasonable sized opening then the whole plate should be removed.

It often occurs that a minor defect is found in a plate and at a time when it is not convenient to stop for repairs; in such an event a "soft" patch is often applied. This consists of a piece of wrought iron carefully fitted to that portion of the boiler plate needing repairs; holes are fitted in both plates and patch, and "patch bolts" provided for them. A thick putty consisting of white and red lead with iron borings or filings in them placed evenly over the inner surface of the patch, which is then tightly

bolted to the boiler plate. This is best but a temporary make-shift and ought never to be regarded as a permanent repair. A mistake is often made of making a patch of thicker metal than that of the shell of the boiler needing it. A moment's reflection ought to show the absurdity of putting on a $\frac{5}{16}$ or $\frac{3}{8}$ patch on an old $\frac{1}{4}$ inch boiler shell; yet it is not so rare as one would imagine. A piece of new iron $\frac{3}{16}''$ thick will, in most cases, be found to be stronger than that portion of a $\frac{1}{4}''$ old plate needing repairs.

Inspection. — A careful external and internal examination of a boiler is to be commended for many reasons. This should be as frequent as possible and thoroughly done; it should include the boiler not only, but all the attachments which affect its working or pressure. Particular attention should be paid to the examination of all braces and stays, safety valve, pressure gauges, water gauges, feed and blow-off apparatus, etc.; these latter refer more particularly to constructive details necessary to proper management and safety. The inspection would obviously be incomplete, did it not include an examination into the causes of "wear and tear," and determine the extent to which it had progressed. Among the several causes which directly tend to rendering a boiler unsafe, may be mentioned the dangerous results occasioned by the overheating of plates, thus changing the structure of the iron from fine granular, or fibrous, to coarse crystalline. This may easily be detected by examination, and will in general be found to occur in such cases where the boilers are too small for the work, are fired too hard, or have a considerable accumulation of scale or sediment in contact with the plates. Blistered plates are almost instantly detected at sight, so also is corrosion, from whatever cause it may have proceeded.

Corrosion of boiler plates. — Iron will corrode rapidly when subjected to the intermittent action of moisture and dryness. Land boilers are less subject to corrosion than marine boilers. The corrosion of a boiler may be either external or internal. Ex-

ternal corrosion may, in general, be easily prevented by carefully caulking all leaks in the boiler; by preventing the dropping of water on the plates, such, for example, as from a leaky joint in the steam pipe or from the safety valve. A leaky roof, by allowing a continual or occasional dropping of water on the top of a boiler, especially if the boiler is not in constant use, would promote external corrosion. Sometimes external corrosion is caused by the use of coal having sulphur in it, and acts in this way: The sulphur passes off from the fire as sulphurous oxide, which often attaches to the sides of a boiler; so long as this is dry no especial mischief is done; but if it comes in contact with a wet plate the sulphurous oxide is converted into sulphuric acid over so much of the surface as the moisture extends; this acid attacks, and will, in time, entirely destroy the boiler plate. Internal corrosion is not so easily accounted for and is very difficult to correct, especially when it occurs above the water line. It is generally believed to be due to the action of acids in the feed water. Marine boilers are especially subject to internal corrosion when used in connection with surface condensers. A few years ago it was generally supposed to be due to galvanic action but that idea is now almost entirely given up. From the fact that boilers using distilled water fed into them from surface condensers are more liable to internal corrosion than other boilers, has led to the theory that it is the *pure* water that does the mischief, and that a water containing in slight degree a scale-forming salt, is to be preferred to water which is absolutely pure. Whatever may be the truth or falsity of this theory, it is a well established fact that distilled water has a most pernicious action on various metals, especially on steel, lead and iron. This action is attributed to its peculiar property, as compared with ordinary water, of dissolving free carbonic acid. One of the worst features in connection with internal corrosion is that its progress cannot be easily traced on account of the boiler being closed while at work. As it does not

usually extend over any very great extent of surface, the ordinary hydraulic test fails to reveal the locality of corroded spots; the hammer test, on the contrary, rarely fails to locate them, if the plates are much thinned by its action.

Testing boilers.—It is the general practice to apply the hydraulic test to all new steam boilers at the place of manufacture, and before shipment. The pressure employed in the test is from one and a half to twice the intended working steam pressure. This test is only valuable in bringing to notice defects which would escape ordinary inspection. It is not to be assumed that it in any way assures good workmanship, or material, or good design, or proper proportions; it simply shows that the boiler being tested is able to withstand this pressure without leaking at the joints, or distorting the shell to an injurious degree. Bad workmanship may often be detected at a glance by an experienced person. The material must be judged by the tensile strength and ductility of the sample tested. The design and proportions are to be judged on constructive grounds, and have little or nothing in common with the hydraulic test. The great majority of buyers of steam boilers have but little knowledge on the subject of tests, and too often conclude that if they have a certified copy of a record showing that a particular boiler withstood a test of say, 150 lbs., it is a good and safe boiler at 75 to 100 lbs. steam pressure. If the boiler is a new one and by a reputable maker, that may be true; if it has been used and put upon the market as a second-hand boiler, it may be anything but safe at half the pressure named. By the hydraulic test, the braces in a boiler may be broken, joints strained so as to make them leak, bolts or pins may be sheared off, or so distorted as to be of little or no service in resisting steam when pressure is on.

Hammer test.—The practice of inspecting boilers by sounding with a hammer is, in many respects, to be commended. It requires some practical experience in order to detect blisters and

the wasting of plates, by sound alone. The hammer test is especially applicable to the thorough inspection of old boilers. It frequently happens in making a test that a blow of the hand hammer will either distort it, or be driven entirely through the plate; and it is just here that the superiority of this method of testing over, or in connection with the hydraulic test, becomes fully apparent. The location of stays, joints and boiler fittings all modify and are apt to mislead the inspector if he depends upon sound alone. There is a certain spring of the hammer and a clear ring indicative of sound plates, which are wanting in plates much corroded or blistered. The presence of scale on the inside of the boiler has a modifying action on the sound of the plate. When a supposed defect is discovered, a hole should be drilled through the sheet by which its thickness may be determined, as well as its condition.

In order to thoroughly inspect a boiler, the inspector should crawl into the boiler (when it is possible to do so) and he should look for pitting and grooving of plates, test all braces, and examine all inlets and outlets.

TOTAL STORED ENERGY OF STEAM BOILERS.

Grate Surface. Sq. Ft.	Heating Surface. Sq. Ft.	Steam Pressure. Lbs.	Rated H. P.	Energy in Foot Pounds Stored in the		
				Water.	Steam.	Total.
15	120	100	10	46605200	676698	47281898
15	875	125	425	64253160	1302431	65555591
20	400	150	35	80572050	2377357	82949407
20	1200	125	600	64452270	1766447	66218718
22	1070	125	525	52561075	1483896	54044971
30	852	75	60	50008790	1022731	51031521
30	1350	125	650	69148790	2135802	71284592
32	768	75	300	71272370	1462430	72734800
36	730	30	60	57570750	709310	58260060
50	1119	75	350	107408340	2316392	109724732
70	2806	100	250	172455270	2108110	174563380
72	1755	30	180	102628410	1643854	104272264
72	2324	30	200	90531490	1570517	92101987
100	3000	100	250	227366000	3513830	230879830

CHAPTER XVII.

USE AND ABUSE OF THE STEAM-BOILER.

Steam-boilers. — A steam-boiler may be defined as a close vessel, in which steam is generated. It may assume an endless variety of forms, and can be constructed of various materials. Since the introduction of steam as a motive power a great variety of boilers have been designed, tried and abandoned; while many others, having little or no merit as steam generators, also have their advocates and are still continued in use. Under such circumstances, it is not surprising that quite a variety of opinions are held on the subject. This difference of opinion relates not only to the form of boiler best adapted to supply the greatest quantity of steam with the least expenditure of fuel, but also to the dimensions or capacity suitable for an engine of a given number of horse-power; and while great improvements have been made in the manufacture of boiler materials within the past fifteen years, yet the number of inferior steam-boilers seem to increase rather than diminish. It would be difficult to assign any reasonable cause for this, except that, of late years, nearly the whole attention of instructors and mechanical engineers has been directed to the improvement and perfection of the steam-engine, and practical engineers, following the example set by the leaders, devote their energies to the same object. This is to be regretted, as the construction and application of the steam-boiler, like the steam-engine, is deserving of the most thorough and scientific study, as on the basis of its employment rest some of the most important interests of civilization. Until quite recently, the idea was very generally entertained that the purely mechanical skill required to enable a person to join

together pieces of metal, and thereby form a steam-tight and water-tight vessel of given dimensions, to be used for the generation of steam to work an engine, was all that was needed; experience has shown, however, that this is but a small portion of the knowledge that should be possessed by persons who turn their attention to the design and construction of steam-boilers, as the knowledge wanted for this end is of a scientific as well as of a mechanical nature. As the boiler is the source of power and the place where the power to be applied is first generated, and also the source from which the most dangerous consequences may arise from neglect or ignorance, it should attract the special attention of the designing and mechanical engineer, as it is well known that from the hour it is set to work, it is acted upon by destroying forces, more or less uncontrollable in their work of destruction. These forces may be distinguished as chemical and mechanical. In most cases they operate independently, though they are frequently found acting conjointly in bringing about the destruction of the boiler, which will be more or less rapid according to circumstances of design, construction, quality of material, management, etc. The causes which most affect the integrity of boilers and limit their usefulness are either inherent in the material, or due to a want of skill in their construction and management; they may be enumerated as follows: —

First, inferior material; second, slag, sand or cinders being rolled into the iron; third, want of lamination in the sheets; fourth, the overstretching of the fiber of the plate on one side and puckering on the other in the process of rolling, to form the circle for the shell of a boiler; fifth, injuries done the plate in the process of punching; sixth, damage induced by the use of the drift-pin; seventh, carelessness in rolling the sheets to form the shell, as a result of which the seams, instead of fitting each other exactly, have in many instances to be drawn together by bolts, which aggravates the evils of expansion and contraction when the

boiler is in use; eighth, injury done the plates by a want of skill in the use of the hammer in the process of hand-riveting; ninth, damage done in the process of calking.

Other causes of deterioration are unequal expansion and contraction, resulting from a want of skill in setting; grooving in the vicinity of the seams; internal and external corrosion; blowing out the boiler when under a high pressure and filling it again with cold water when hot; allowing the fire to burn too rapidly after starting, when the boiler is cold; ignorance of the use of the pick in the process of scaling and cleaning; incapacity of the safety-valve; excessive firing; urging or taxing the boiler beyond its safe and easy working capacity; allowing the water to become low, and thus causing undue expansion; deposits of scale accumulating on the parts exposed to the direct action of the fire, thereby burning or crystallizing the sheets or shell; wasting of the material by leakage and corrosion; bad design and construction of the different parts; inferior workmanship and ignorance in the care and management. All these tend with unerring certainty to limit the age and safety of steam boilers. On account of want of skill on the part of the designer and avarice on the part of the manufacturer, or perhaps both reasons, boilers are sometimes so constructed as to bring a riveted seam directly over the fire, the result of which is that in consequence of one lap covering the other, the water is prevented from getting to the one nearest the fire, for which reason the lap nearest the fire becomes hotter and expands to a much greater extent than any other part of the plate; and its constant unequal expansion and contraction, as the boiler becomes alternately hot and cold, inevitably results in a crack. Such blunders are aggravated by the scale and sediment being retained on the inside, between the heads of the rivets, which should be properly removed in cleaning.

The tendency of manufacturers to work boilers beyond their capacity, especially when business is driving, is too great in this

country; and no doubt many boiler explosions may be attributed to this cause. Boilers are bought, adapted to the wants of the manufactory at the time, but, as business increases, machinery is added to supply the demand for goods, until the engine is overtasked, the boiler strained and rendered positively dangerous. Then again, it not unfrequently occurs that engines in manufactories are taken out and replaced by those of increased power, while the boilers used with the old engine are retained in place, with more or less cleaning and patching, as the case may require. Now, it is evident to any practical mind that boilers constructed for a twenty horse power engine are ill adapted to an engine of forty horse power, more especially if those boilers have been used for a number of years. In order to supply sufficient steam for the new engine, with a cylinder of increased capacity, the boiler must be worked beyond its safe working pressure, consequently excessive heating and pressure greatly weaken it and endanger the lives of those employed in the vicinity.

The danger and impracticability of using boilers with too limited steam room may be explained thus: Suppose the entire steam room in a boiler to be six cubic feet, and the contents of the cylinder which it supplies to be two cubic feet; then at each stroke of the piston one-third of all the steam in the boilers is discharged, and consequently, one-third of the pressure on the surface of the water before that stroke is relieved; hence, it will be seen that excessive fires must be kept up in order to generate steam of sufficiently high temperature and pressure to supply the demand. The result is that the boilers are strained and burned. Such economy in boiler power is exceedingly expensive in fuel, to say nothing of the danger. Excessive firing distorts the fire-sheets, causing leakage, undue and unequal expansion and contraction, fractures, and the consequent evils arising from external corrosion. Excessive pressure arises generally from a desire on the part of the steam-user to make a boiler do double the work for

which it was originally intended. A boiler that is constructed to work safely at from fifty to sixty pounds was never intended to run at eighty and ninety pounds; more especially if it had been in use for several years. Boilers deteriorated by age should have their pressure decreased, rather than increased.

One of the first things that should be done in manufacturing establishments would be to provide sufficient boiler power and, in order to do this, the work to be done ought to be accurately calculated and the engine and boilers adapted to the results of this calculation. Steam users themselves are frequently to blame for the annoyances and dangers arising from unsafe boilers and those of insufficient capacity. From motives of false economy they are too easily swayed in favor of the cheaper article, simply because it is cheap, when they should consider they are purchasing an article which, of almost all others, should be made in the most thorough manner and of the best material. In view of the fearful explosions that occur from time to time, every steam user should secure for his use the best and safest. The object of a few dollars as between the work of a good, responsible maker and that of an irresponsible one, should not for one moment be entertained.

It is very bad policy for steam-users to advertise for estimates for steam-boilers, or to inform all the boilermakers in the town or city that a boiler or boilers to supply steam for an engine of a certain size is needed, because in this way steam-users frequently find themselves in the hands of needy persons, who, in their anxiety to get an order, will sometimes ask less for a boiler than they can actually make it for; consequently, they have to cheat in the material, in the workmanship, in the heating surface and in the fittings. As a result, the boiler is not only a continual source of annoyance, but, in many instances, an actual source of danger. The most prudent course, and in fact the only one that may be expected to give satisfaction, is to contract with some responsible

manufacturer that has an established reputation for honesty, capability and fair dealing, and who will not allow himself to be brought in competition with irresponsible parties for the purpose of selling a boiler. There are thousands of boilers designed, constructed and set up in such a manner as to render it utterly impossible to examine, clean or repair them. Generally, in such cases, in consequence of imperfect circulation, the water is expelled from the surface of the iron at the points where the extreme heat from the furnace impinges, and, as a result, the plates become overheated and bulge outward, which aggravates the evil, as the hollow formed by the bulge becomes a receptacle for scale and sediment. By continued overheating, the parts become crystallized and either crack or blister; this, if not attended to and remedied, will eventually end in the destruction of the boiler. Many boilers, to all appearance well made and of good material, give considerable trouble by leakage and fracture, owing to the severe strains of unequal expansion and contraction induced by the rigid construction, the result of a want of skill in the original design.

DESIGN OF STEAM-BOILERS.

It has become a general assertion on the part of writers on the steam-boiler that the most important object to be attained in its design and arrangement is thorough combustion of the fuel. This is only partially true as there are other conditions equally important, among which are strength, durability, safety, economy and adaptability to the particular circumstances under which it is to be used. However complete the combustion may be, unless its products can be easily and rapidly transferred to the water, and unless the means of escape of the steam from the surfaces on which it is generated is easy and direct, the boiler will fail to produce satisfactory results, either in point of durability or economy of fuel.

Strength means the power to sustain the internal pressure to which the boiler may be subjected in ordinary use, and under careful and intelligent management. To secure durability, the material must be capable of resisting the chemical action of the minerals contained in the water, and the boiler ought to be designed so as to procure the least strain under the highest state of expansion to which it may be subjected — be so constructed that all the parts will be subjected to an equal expansion, contraction, push, pull and strain, and be intelligently and thoroughly cared for after being put in use. These objects, however, can only be obtained by the aid of a knowledge of the principles of mechanics, the strength and resistance of materials, the laws of expansion and contraction, the action of heat on bodies, etc. The economy of a steam boiler is influenced by the following conditions: cost and quantity of the material, design, character of the workmanship employed in its construction, space occupied, capability of the material to resist the chemical action of the ingredients contained in the water, the facilities it affords for the transmission of the heat from the furnace to the water, etc. The safety of any structure depends on the designer's knowledge of the principles of mechanics, the resistance of materials and the action of bodies as influenced by the elements to which they are exposed; and in the case of steam boilers, the safety depends on the judgment of the designer, the quality of the material, the character of the workmanship and the skill employed in the management. Safety is said to be incompatible with economy, but this is undoubtedly a mistake, as an intelligent economy includes permanence and seeks durability. Adaptability to the peculiar purposes for which they are to be used is one of the first objects to be sought for in the design and construction of any class of machines, vessels or instruments, and it is undoubtedly this that gave rise to the great variety of designs, forms and modifications of steam boilers in use at the present day, which are, with very

few exceptions, the result of thought, study, investigation and experiment.

FORMS OF STEAM-BOILERS.

According to the well-known law of hydrostatics, the pressure of steam in a close cylindrical vessel is exerted equally in all directions. In acting against the circumference of a cylinder, the pressure must, therefore, be regarded as radiating from the axis, and exerting a uniform tensional strain throughout the inclosing material.

Familiarity with steam machinery, more especially with boilers, is apt to beget a confidence in the ignorant which is not founded on a knowledge of the dangers by which they are continually surrounded; while contact with steam, and a thoroughly elementary knowledge of its constituents, theory and action, only incline the intelligent engineer and fireman to be more cautious and energetic in the discharge of their duties. Many regard steam as an incomprehensible mystery; and although they may employ it as a power to accomplish work, know little of its character or capabilities. Steam may be managed by common sense rules as well as any other power; but if the laws which regulate its use are violated, it reports itself, and often in louder tones than is pleasant. If steam-boilers in general were better cared for than they are, their working age might be greatly increased. Deposits of incrustation, small leaks and slight corrosion, are too often neglected as matters of little consequence, but they are the forerunners of expensive repairs, delay and disaster.

SETTING STEAM-BOILERS.

While engineers differ very much in opinion respecting the best manner of setting boilers, they all readily allow that the results obtained, as regards economy of fuel and the generation of steam,

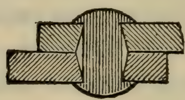
depend in a great measure on the arrangement of the setting. Particularly is this the case with horizontal tubular boilers, and there have been numerous plans introduced to obtain a maximum of steam with a minimum of fuel. Some of the most practical designs and best laid plans are frequently rendered useless for want of knowledge on the part of those whose duty it is to execute or carry them out. This has perhaps been more frequently the case as regards the setting of steam boilers than any other class of machines, as it is customary for owners of steam boilers to depend too much on the knowledge of masons and bricklayers; consequently, a great many blunders have been made which necessitated changes in the size of gratebars, alteration of brickwork, alteration of flues, chimney, etc., with a list of other annoyances, such as insufficiency of steam, poor draught, or something else. In setting or putting in boilers, all the surface possible should be exposed to the action of the heat of the fire, not only that the heat may be thus completely absorbed, but that a more equal expansion and contraction of the structure may be obtained. Long boilers are often hung by means of loops riveted to the top of them and connected to crossbeams and arches resting on masonry above them, by means of hangers. This is a very mischievous arrangement, unless turn-buckles, or some other contrivance, are used to maintain a regular strain on all the hangers, as long boilers exposed to excessive heat are apt to lengthen on the lower side and relieve the end hangers of any weight; consequently, the whole strain is transmitted to the central hanger, which has a tendency to draw the boiler out of shape — in many instances inducing excessive leakage, rupture, and, eventually, explosion.

DEFECTS IN THE CONSTRUCTION OF STEAM-BOILERS.

The following cuts illustrate some of the mechanical defects that impair the strength and limit the safety and durability of

steam boilers. All punched holes are conical, and unless the sheets are reversed after being punched, so as to bring the small sides of the holes together, it will be impossible to fill them with the rivets. Fig. 251 shows the position of the rivet in the hole without the sheets being reversed; and it will be observed that, as very little of the rivet bears against the material, the expansion and contraction of the boiler have a tendency to work it loose. It is apparent that such a seam would not possess over one-third the strength that it would if the holes in the sheets

Fig. 251.



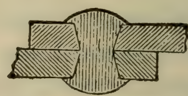
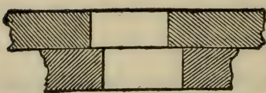


Fig. 252.

Fig. 253.



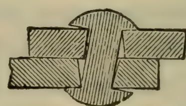


Fig. 254.

Fig. 255.

Fig. 256.

Showing positions of rivet in rivet hole.

were reversed and thoroughly filled with the rivet, as shown in Fig. 252. Fig. 253 represents what is known in boilermaking as a blind hole, which means that the holes do not come opposite each other when the seams are placed together for the purpose of riveting. Fig. 254 shows the position of the rivet in the blind hole after being driven. It will be observed that the heads of the rivet, in consequence of its oblique position in the hole, bear only on one side, and that even the bearing is very limited, and through the expansion and contraction of the boiler, is liable to

work loose and become leaky. Such a seam would be actually weaker than that presented in Fig. 251. Fig. 255 shows the metal distressed and puckered on each side of the blind hole in the sheets, which is the result of efforts on the part of the boiler-maker, by the use of the drift-pin, to make the holes correspond for the purpose of inserting the rivet. Fig. 256 shows the metal broken through by the same means. Now, it will be observed that nearly all the above defects are the result of ignorance and carelessness, showing a want of skill in laying out the work, as well as a want of proper appliances for that purpose. The evils arising from such defects are greatly aggravated by the fact that they are all concealed, frequently defying the closest scrutiny, and are only revealed by those forces which unceasingly act on boilers when in use. Such pernicious mechanical blunders ought to be condemned, as they are always the forerunners of destruction and death. There can be no reason why boilers should not be constructed with the same degree of accuracy, judgment and skill as is considered so essential for all other classes of machinery.

IMPROVEMENTS IN STEAM BOILERS.

Until quite recently the steam boiler has undergone very little improvement. This arose, perhaps, from the fact that men of intelligence and mechanical genius directed their thoughts and labors to something more inviting and less laborious than the construction of steam boilers. Consequently, that branch of mechanics was left almost entirely to a class of men that had not the genius to rise in their profession or improve much in anything they attempted. As a result ignorance, stupidity and a kind of brute force were the predominant requirements in the construction of the steam boiler ; but within the past few years this state of things has been changed, as some very important improvements have been made, not only in the manufacture of the material of which boilers are made, but also in the mode of constructing

them. The imposing, powerful and accurate boiler machinery in use at the present time is an evidence that the attention of eminent mechanics and manufacturers is directed to the steam boiler, and that in the future its improvement will keep pace with that of the steam engine.

Boiler plate is now rolled of sufficient dimensions to form the rings for boilers of any diameter with only one seam, obviating the necessity of bringing riveted seams in contact with the fire, as was usually the case in former times. In the manner of laying off the holes for the rivets, accurate steel gauges have taken the place of the old-fashioned wooden templet, thereby removing the evils induced by blind holes, and obviating the necessity of using the drift-pin. So, also, in the method of bending the sheets to form the requisite circle — with a better class of machinery, the work is now more accurately performed. The old process of chipping is, in nearly all the large boilershops, superseded by planing the bevels on the edge of the sheet, preparatory to calking. Recent improvements in "calking" have resulted in perfect immunity from the injuries formerly inflicted on boilers in that process. In most establishments of any repute in this country, riveting is done by machinery, which is (as is well known to all intelligent mechanics) very much superior to hand riveting. It is only small shops that enter into rivalry to secure orders and build cheap boilers, using poor material and an inferior quality of mechanical skill, that use the same old crude appliances — in many cases the merest makeshifts — that were in use a quarter of a century ago, and constructed without regard to any of the rules of design that are considered so essential in appliances for the construction of all other classes of machinery. Every engineer should inform himself on the subject of the safe working pressure of boilers, and when he finds the limit of safety has been reached, he should promptly inform his employer and use his influence to have the boiler worked within the bounds of safety.

To find the heating surface of a water tube boiler: —

Rule. — Add the combined outside area of the tubes in square feet to one-half the area of the shell of the steam drum in square feet and the sum will give the total heating surface.

Example 1. — What is the heating surface of a water tube boiler having fifty tubes, each three inches outside diameter and fifteen feet long, and the steam drum thirty-two inches in diameter and fifteen feet long?

Operation. — 3×3.1416 equals 9.4248 inches, the circumference of one tube. 15×12 equals 180 inches the length of one tube. $\frac{9.4248 \times 180}{144}$ equals 11.781 square feet in one tube, and 11.781×50 equals 589.05 square feet of heating surface in fifty tubes. Then, $\frac{32 \times 3.1416}{12}$ equals 8.3776 linear feet the circumference of the steam drum and 8.3776×15 equals 125.664 square feet of heating surface in steam drum, and $\frac{125.664}{2}$ equals 62.832 square feet, half the heating surface of steam drum.

Then, 589.05 plus 62.832 equals 651.882 square feet, the total heating surface. Answer.

STRENGTH OF RIVETED SEAMS.

The strength of a riveted seam depends very much upon the arrangement and proportion of the rivets; but with the best design and construction, the seams are always weaker than the solid plate, as it is always necessary to cut away a part of the plate for the rivet holes, which weakens the plate in three ways: 1st, by lessening the amount of material to resist the strains; 2d, by weakening that left between the holes; 3d, by disturbing the uniformity of the distribution of the strains.

COMPARATIVE STRENGTH OF SINGLE AND DOUBLE RIVETED SEAMS.

On comparing the strength of plates with riveted joints, it will be necessary to examine the sectional areas taken in a line through the rivet holes, with the section of the plates themselves. It is obvious that in perforating a line of holes along the edge of a plate, we must reduce its strength. It is also clear that the plate so perforated will be to the plate itself nearly as the areas of their respective sections, with a small deduction for the irregularities of the pressure of the rivets upon the plate; or, in other words, the joint will be reduced in strength somewhat more than in the ratio of its section through that line to the solid section of the plate. It is also evident that the rivets cannot add to the strength of the plates, their object being to keep the two surfaces of the lap in contact. When this great deterioration of strength at the joint is taken into account, it cannot but be of the greatest importance that in structures subject to such violent strains as boilers, the strongest method of riveting should be adopted. To ascertain this, a long series of experiments was undertaken by Mr. Fairbairn. There are two kinds of lap joints, single and double riveted. In the early days of steam-boiler construction, the former were almost universally employed; but the greater strength of the latter has since led to their general adoption for all boilers intended to sustain a high steam pressure. A riveted joint generally gives way either by shearing off the rivets in the middle of their length, or by tearing through one of the plates in the line of the rivets.

In a perfect joint, the rivets should be on the point of shearing just as the plates were about to tear; but, in practice, the rivets are usually made slightly too strong. Hence, it is an established rule to employ a certain number of rivets per linear foot, which for ordinary diameters and average thickness of plate, are about

six per foot or two inches from center to center; for larger diameters and heavier iron, the distance between the centers is generally increased to, say $2\frac{1}{8}$ or $2\frac{1}{4}$ inches; but in such cases it is also necessary to increase the diameter of the rivet, for while $\frac{5}{8}$, or even $\frac{1}{2}$ inch rivets will answer for small diameters and light plate, with large diameters and heavy plate, experience has shown it to be necessary to use $\frac{3}{4}$ to $\frac{7}{8}$ rivets. If these are placed in a single row, the rivet holes so nearly approach each other that the strength of the plates is much reduced; but if they are arranged in two lines, a greater number may be used, more space left between the holes and greater strength aud stiffness imparted to the plates at the joint. Taking the value of the plate before being punched, at 100, by punching the plate it loses 44 per cent of its strength; and, as a result, single-riveted seams are equal to 56 per cent, and double-riveted seams to 70 per cent of the original strength of the plate. It has been shown by very extensive experiments at the Brooklyn Navy Yard, and also at the Stevens Institute of Technology, Hoboken, N. J., that double-riveted seams are from 16 to 20 per cent stronger than single-riveted seams — the material and workmanship being the same in both cases:

Taking the strength of the plate at	100
The strength of the double-riveted joint would then be . .	70
The strength of the single-riveted would be	56

To find the thickness of plates for the shell of a cylindrical boiler for a required safe working pressure in pounds per square inch: —

Rule. — Multiply the required pressure per square inch by the radius of the shell in inches, and by the constant number 6 for single riveted side seams, and divide the last product by the tensile strength of the plates. For double riveted side seams use the constant number 5 instead of 6.

Example 1. — What should be the thickness of plates for a boiler 60 inches in diameter, with single riveted side seams, for a work-

ing pressure of 125 pounds per square inch, the tensile strength of the plates being 60,000 pounds per square inch?

Operation. — $\frac{125 \times 30 \times 6}{60,000}$ equals .375 or 3/8 in. Answer.

Example 2. — What should be the thickness of plates for a boiler 60 inches diameter, with double riveted side seams, for a working pressure of 150 pounds per square inch, the tensile strength of plates being 60,000 pounds per square inch.

Operation. — $\frac{150 \times 30 \times 5}{60,000}$ equals .375 or 3/8 in. Answer.

The following formulas, equivalent to those of the British Board of Trade, are given for the determination of the pitch, distance between rows of rivets, diagonal pitch, maximum pitch, and distance from centers of rivets to edge of lap of single and double riveted lap joints, for both iron and steel boilers: —

Let p = greatest pitch of rivets, in inches;
n = number of rivets, in one pitch;
p_d = diagonal pitch, in inches;
d = diameter of rivets, in inches;
T = thickness of plate, in inches;
V = distance between rows of rivets, in inches;
E = distance from edge of plate to center of rivet, in inches.

TO DETERMINE THE PITCH.

Iron plates and iron rivets —

$$p = \frac{d^2 \times .7854 \times n}{T} + d.$$

Example: First, for single-riveted joint —

Given, thickness of plate $(T) = \frac{1}{2}$ inch, diameter of rivet $(d) = \frac{7}{8}$ inch. In this case, $n = 1$. Required, the pitch.

Substituting in formula, and performing operation indicated.

$$\text{Pitch} = \frac{(\frac{7}{8})^2 \times .7854 \times 1}{\frac{1}{2}} + \frac{7}{8} = 2.077 \text{ inches.}$$

For double-riveted joint —

Given, $t = \frac{1}{2}$ inch, and $d = \frac{13}{16}$ inch. In this case, $n = 2$. Then —

$$\text{Pitch} = \frac{(\frac{13}{16})^2 \times .7854 \times 2}{\frac{1}{2}} + \frac{13}{16} = 2.886 \text{ inches.}$$

For *steel* plates and steel rivets: —

$$p = \frac{23 \times d^2 \times n}{28 \times T} + d.$$

Example, for single-riveted joint: Given, thickness of plate $= \frac{1}{2}$ inch, diameter of rivet $= \frac{15}{16}$ inch. In this case, $n = 1$. Then —

$$\text{Pitch} = \frac{23 \times (\frac{15}{16})^2 \times .7854 \times 1}{28 \times \frac{1}{2}} + \frac{15}{16} = 2.071 \text{ inches.}$$

Example, for double-riveted joint: Given, thickness of plate $= \frac{1}{2}$ inch, diameter of rivet $= \frac{7}{8}$ inch. $n = 2$. Then —

$$\text{Pitch} = \frac{23 \times (\frac{7}{8})^2 \times .7854 \times 2}{28 \times \frac{1}{2}} + \frac{7}{8} = 2.85 \text{ inches.}$$

FOR DISTANCE FROM CENTER OF RIVET TO EDGE OF LAP.

$$E = \frac{3 \times d}{2}.$$

Example: Given, diameter of rivet $(d) = \frac{7}{8}$ inch; required, the distance from center of rivet to edge of plate.

$$E = \frac{3 \times \frac{7}{8}}{2} = 1.312 \text{ inches,}$$

for single or double riveted lap joint.

FOR DISTANCE BETWEEN ROWS OF RIVETS.

The distance between lines of centers of rows of rivets for double, chain-riveted joints (V) should not be less than twice the diameter of rivet, but it is more desirable that V should not be less than $\frac{4d + 1}{2}$.

Example under latter formula: Given, diameter of rivet $= \frac{7}{8}$ inch, then —

$$V = \frac{(4 \times \frac{7}{8}) + 1}{2} = 2.25 \text{ inches.}$$

For ordinary, double, zigzag-riveted joints,

$$V = \frac{\sqrt{(11p + 4d)\ (p + 4d)}}{10}.$$

Example: Given, pitch $= 2.85$ inches, and diameter of rivet $= \frac{7}{8}$ inch, then —

$$V = \frac{\sqrt{(11 \times 2.85 + 4 \times \frac{7}{8})\ (2.85 + 4 \times \frac{7}{8})}}{10} = 1.487 \text{ inches.}$$

DIAGONAL PITCH.

For double, zigzag-riveted lap joint. Iron and steel.

$$p_d = \frac{6p + 4d}{10}.$$

Example: Given, pitch $= 2.85$ inches, and $d = \frac{7}{8}$ inch, then —

$$p_d = \frac{(6 \times 2.85) + (4 \times \frac{7}{8})}{10} = 2.06 \text{ inches.}$$

MAXIMUM PITCHES FOR RIVETED LAP JOINTS.

For single-riveted lap joints, maximum pitch $= (1.31 \times T) + 1\frac{5}{8}$.

For double-riveted lap joints, maximum pitch $= (2.62 \times T) + 1\frac{5}{8}$.

Example: Given a thickness of plate $= \frac{1}{2}$ inch, required, the maximum pitch allowable.

For single-riveted lap joint, maximum pitch $= (1.31 \times \frac{1}{2}) + 1\frac{5}{8} = 2.28$ inches.

For double-riveted lap joint, maximum pitch $= (2.62 \times \frac{1}{2}) + 1\frac{5}{8} = 2.935$ inches.

The following tables, taken from the handbook of Thomas W. Traill, entitled "Boilers, Marine and Land, their Construction

and Strength," may be taken for use in single and double riveted joints, as approximating the formulas of the British Board of Trade for such joints: —

IRON PLATES AND IRON RIVETS.

DOUBLE-RIVETED LAP JOINTS.

Thickness of plates.	Diameter of rivets.	Pitch of rivets.	Center of rivets to edge of plates.	Distance between rows of rivets.	
				Zigzag riveting.	Chain riveting.
T	d	p	E	V	V
$\frac{5}{16}$	$\frac{5}{8}$	2.272	.937	1.145	1.750
$\frac{11}{32}$	$\frac{21}{32}$	2.386	.984	1.202	1.812
$\frac{3}{8}$	$\frac{11}{16}$	2.500	1.031	1.260	1.875
$\frac{13}{32}$	$\frac{23}{32}$	2.613	1.078	1.317	1.937
$\frac{7}{16}$	$\frac{3}{4}$	2.727	1.125	1.374	2.000
$\frac{15}{32}$	$\frac{25}{32}$	2.826	1.171	1.426	2.062
$\frac{1}{2}$	$\frac{13}{16}$	2.886	1.218	1.465	2.125
$\frac{17}{32}$	$\frac{27}{32}$	2.948	1.265	1.504	2.187
$\frac{9}{16}$	$\frac{7}{8}$	3.013	1.312	1.544	2.250
$\frac{19}{32}$	$\frac{29}{32}$	3.079	1.359	1.585	2.312
$\frac{5}{8}$	$\frac{15}{16}$	3.146	1.406	1.626	2.375
$\frac{21}{32}$	$\frac{31}{32}$	3.215	1.453	1.667	2.437
$\frac{11}{16}$	1	3.284	1.500	1.709	2.500
$\frac{23}{32}$	$1\frac{1}{32}$	3.355	1.546	1.751	2.562
$\frac{3}{4}$	$1\frac{1}{16}$	3.426	1.593	1.794	2.625
$\frac{25}{32}$	$1\frac{3}{32}$	3.498	1.640	1.836	2.687
$\frac{13}{16}$	$1\frac{1}{8}$	3.571	1.687	1.879	2.750
$\frac{27}{32}$	$1\frac{5}{32}$	3.645	1.734	1.923	2.812
$\frac{7}{8}$	$1\frac{3}{16}$	3.718	1.781	1.966	2.875
$\frac{29}{32}$	$1\frac{7}{32}$	3.793	1.828	2.009	2.937
$\frac{15}{16}$	$1\frac{1}{4}$	3.867	1.875	2.053	3.000
$\frac{31}{32}$	$1\frac{9}{32}$	3.942	1.921	2.096	3.062
1	$1\frac{5}{16}$	4.018	1.968	2.140	3.125

On the following page, Fig. 257 shows a zigzag, and Fig. 258 a chain riveted joint.

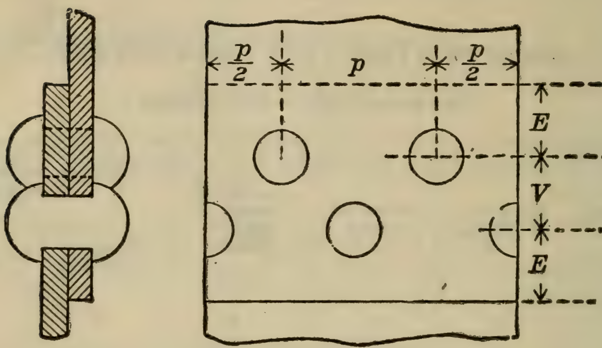


Fig. 257. Zigzag riveted joint.

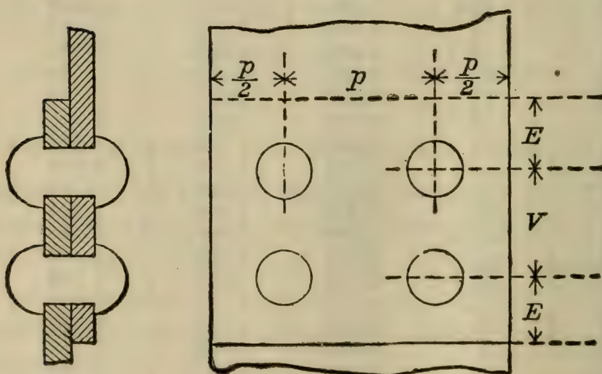
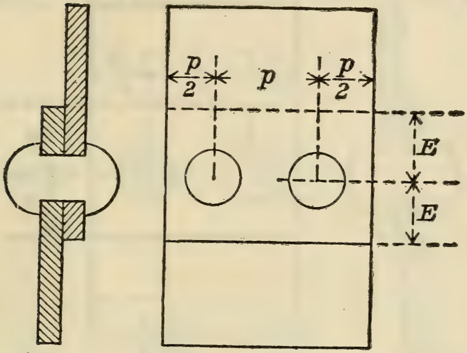


Fig. 258. Chain riveted joint.

IRON PLATES AND IRON RIVETS.

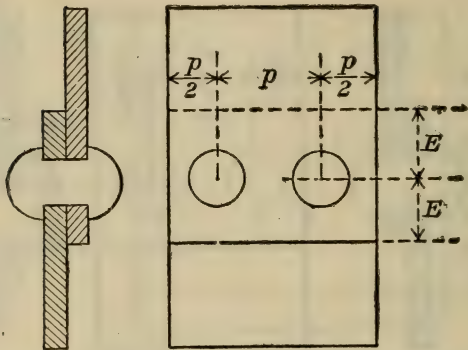
SINGLE-RIVETED LAP JOINTS.



Thickness of plates.	Diameter of rivets.	Pitch of rivets.	Center of rivets to edge of plates.
T	d	p	E
$\frac{1}{4}$	$\frac{5}{8}$	1.524	.937
$\frac{9}{32}$	$\frac{21}{32}$	1.600	.984
$\frac{5}{16}$	$\frac{11}{16}$	1.676	1.031
$\frac{11}{32}$	$\frac{23}{32}$	1.753	1.078
$\frac{3}{8}$	$\frac{3}{4}$	1.829	1.125
$\frac{13}{32}$	$\frac{25}{32}$	1.905	1.171
$\frac{7}{16}$	$\frac{13}{16}$	1.981	1.218
$\frac{15}{32}$	$\frac{27}{32}$	2.036	1.265
$\frac{1}{2}$	$\frac{7}{8}$	2.077	1.312
$\frac{17}{32}$	$\frac{29}{32}$	2.120	1.359
$\frac{9}{16}$	$\frac{15}{16}$	2.164	1.406
$\frac{19}{32}$	$\frac{31}{32}$	2.210	1.453
$\frac{5}{8}$	1	2.256	1.500
$\frac{21}{32}$	$1\frac{1}{32}$	2.304	1.546
$\frac{11}{16}$	$1\frac{1}{16}$	2.352	1.593
$\frac{23}{32}$	$1\frac{3}{32}$	2.400	1.640
$\frac{3}{4}$	$1\frac{1}{8}$	2.450	1.687
$\frac{25}{32}$	$1\frac{5}{32}$	2.500	1.734
$\frac{13}{16}$	$1\frac{3}{16}$	2.550	1.781
$\frac{27}{32}$	$1\frac{7}{32}$	2.601	1.828
$\frac{7}{8}$	$1\frac{1}{4}$	2.652	1.875
$\frac{29}{32}$	$1\frac{9}{32}$	2.703	1.921
$\frac{15}{16}$	$1\frac{5}{16}$	2.755	1.968

STEEL PLATE AND STEEL RIVETS.

SINGLE-RIVETED LAP JOINTS.



Thickness of plates.	Diameter of rivets.	Pitch of rivets.	Center of rivets to edge of plates.
T	d	P	E
$\frac{1}{4}$	$\frac{11}{16}$	1.562	1.031
$\frac{9}{32}$	$\frac{23}{32}$	1.633	1.078
$\frac{5}{16}$	$\frac{3}{4}$	1.704	1.125
$\frac{11}{32}$	$\frac{25}{32}$	1.775	1.171
$\frac{3}{8}$	$\frac{13}{16}$	1.846	1.218
$\frac{13}{32}$	$\frac{27}{32}$	1.917	1.265
$\frac{7}{16}$	$\frac{7}{8}$	1.988	1.312
$\frac{15}{32}$	$\frac{29}{32}$	2.036	1.359
$\frac{1}{2}$	$\frac{15}{16}$	2.071	1.406
$\frac{17}{32}$	$\frac{31}{32}$	2.108	1.453
$\frac{9}{16}$	1	2.146	1.500
$\frac{19}{32}$	$1\frac{1}{32}$	2.186	1.546
$\frac{5}{8}$	$1\frac{1}{16}$	2.227	1.593
$\frac{21}{32}$	$1\frac{3}{32}$	2.269	1.640
$\frac{11}{16}$	$1\frac{1}{8}$	2.312	1.687
$\frac{23}{32}$	$1\frac{5}{32}$	2.356	1.734
$\frac{3}{4}$	$1\frac{3}{16}$	2.400	1.781
$\frac{25}{32}$	$1\frac{7}{32}$	2.445	1.828
$\frac{13}{16}$	$1\frac{1}{4}$	2.500	1.875
$\frac{27}{32}$	$1\frac{9}{32}$	2.562	1.921
$\frac{7}{8}$	$1\frac{5}{16}$	2.623	1.968
$\frac{29}{32}$	$1\frac{11}{32}$	2.687	2.015
$\frac{15}{16}$	$1\frac{3}{8}$	2.750	2.062

STEEL PLATE AND STEEL RIVETS.

DOUBLE-RIVETED LAP JOINTS.

Thickness of plates.	Diameter of rivets.	Pitch of rivets.	Center of rivets to edge of plates.	Distance between rows of rivets.	
				Zigzag riveting.	Chain riveting.
T	*d*	*P*	*E*	*V*	*V*
$\frac{5}{16}$	$\frac{11}{16}$	2.291	1.031	1.187	1.875
$\frac{11}{32}$	$\frac{23}{32}$	2.395	1.078	1.240	1.937
$\frac{3}{8}$	$\frac{3}{4}$	2.500	1.125	1.295	2.000
$\frac{13}{32}$	$\frac{25}{32}$	2.604	1.171	1.349	2.062
$\frac{7}{16}$	$\frac{13}{16}$	2.708	1.218	1.403	2.125
$\frac{15}{32}$	$\frac{27}{32}$	2.803	1.265	1.453	2.187
$\frac{1}{2}$	$\frac{7}{8}$	2.850	1.312	1.487	2.250
$\frac{17}{32}$	$\frac{29}{32}$	2.900	1.359	1.522	2.312
$\frac{9}{16}$	$\frac{15}{16}$	2.953	1.406	1.558	2.375
$\frac{19}{32}$	$\frac{31}{32}$	3.008	1.453	1.595	2.437
$\frac{5}{8}$	1	3.064	1.500	1.631	2.500
$\frac{21}{32}$	$1\frac{1}{32}$	3.122	1.546	1.669	2.562
$\frac{11}{16}$	$1\frac{1}{16}$	3.181	1.593	1.707	2.625
$\frac{23}{32}$	$1\frac{3}{32}$	3.241	1.640	1.745	2.687
$\frac{3}{4}$	$1\frac{1}{8}$	3.302	1.684	1.784	2.750
$\frac{25}{32}$	$1\frac{5}{32}$	3.364	1.734	1.823	2.812
$\frac{13}{16}$	$1\frac{3}{16}$	3.427	1.781	1.863	2.375
$\frac{27}{32}$	$1\frac{7}{32}$	3.490	1.828	1.902	2.937
$\frac{7}{8}$	$1\frac{1}{4}$	3.554	1.875	1.942	3.000
$\frac{29}{32}$	$1\frac{9}{32}$	3.618	1.921	1.981	3.062
$\frac{15}{16}$	$1\frac{5}{16}$	3.683	1.968	2.021	3.125
$\frac{31}{32}$	$1\frac{11}{32}$	3.748	2.015	2.061	3.187
1	$1\frac{3}{8}$	3.814	2.062	2.102	3.250

On the following page Fig. 259 shows a zigzag riveted joint and Fig. 260 a chain riveted joint with steel plate and steel rivets.

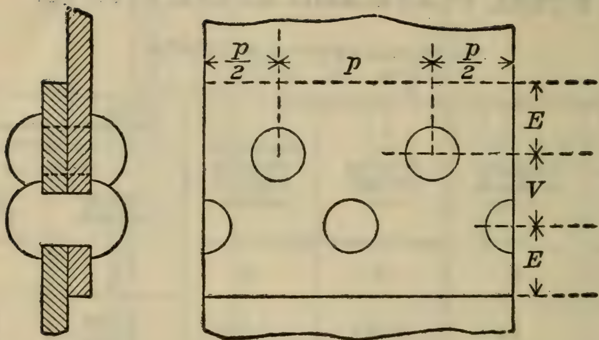


Fig. 259. Zigzag riveted joint.

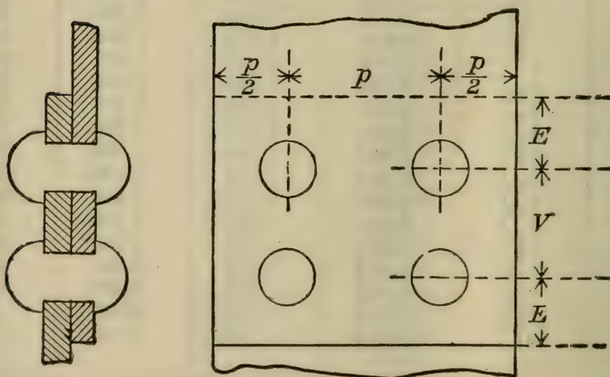


Fig. 260. Chain riveted joint.

STRENGTH OF STAYED AND FLAT BOILER SURFACES.

The sheets that form the sides of fire-boxes are necessarily exposed to a vast pressure, therefore, some expedient has to be devised to prevent the metal at these parts from bulging out. Stay-bolts are generally placed at a distance of $4\frac{1}{2}$ inches from center to center, all over the surface of fire-boxes, and thus the expansion or bulging of one side is prevented by the stiffness or rigidity of the other. Now, in an arrangement of this kind, it becomes necessary to pay considerable attention to the tensile strength of the stay-bolts employed for the above purpose, since the ultimate strength of this part of the boiler is now transferred to them, it being impossible that the boiler plates should give way unless the stay-bolts break in the first instance. Accordingly, the experiments that have been made by way of test of the strength of stay-bolts, possess the greatest interest for the practical engineer. Mr. Fairburn's experiments are particularly valuable. He constructed two flat boxes, 22 inches square. The top and bottom plates of one were formed of $\frac{1}{2}$ inch copper, and of the other, $\frac{3}{8}$ inch iron. There was a $2\frac{1}{2}$ inch water-space to each, with $\frac{13}{16}$ inch iron-stays screwed into the plates and riveted on the ends. In the first box the stays were placed five inches from center to center, and the two boxes tested by hydraulic pressure. In the copper box, the sides commenced to bulge at 450 lbs. pressure to the sq. in.; and at 815 lbs. pressure to the sq. in. the box burst, by drawing the head of one of the stays through the copper plate. In the second box, the stays were placed at 4-inch centers; the bulging commenced at 515 lbs. pressure to the sq. in. The pressure was continually augmented up to 1,600 lbs. The bulging between the rivets at that pressure was one-third of an inch; but still no part of the iron gave way. At 1,625 lbs. pressure the box burst, and in precisely the same way as in the first experiment — one of the stays drawing through the

iron plate and stripping the thread in plate. These experiments prove a number of facts of great value and importance to the engineer. In the first place, they show that with regard to iron stay-bolts, their tensile strength is at least equal to the grip of the plate.

The grip of the copper bolt is evidently less. As each stay, in the first case, bore the pressure on an area of 5 x 5 = 25 square inches, and in the second on an area 4 x 4 = 16 sq. inches, the total strains borne by each stay were, for the first, 815 x 25 = 20,375 pounds on each stay; and for the second, 1,625 x 16 = 26,000 lbs. on each stay. These strains were less, however, than the tensile strength of the stays, which would be about 28,000 lbs. The properly stayed surfaces are the strongest part of boilers, when kept in good repair.

BOILER STAYS.

Advantage is usually taken of the self-supporting property of the cylinder and sphere, which enables them, in most cases, to be made sufficiently strong without the aid of stays or other support. But the absence of this self-sustaining property in flat surfaces necessitates their being strengthened by stays or other means. Even where a flat or slightly dished surface possesses sufficient strength to resist the actual pressure to which it is subjected, it is yet necessary to apply stays to provide against undue deflection or distortion, which is liable to take place to an inconvenient degree, or to result in grooving, long before the strength of plates or their attachments is seriously taxed. Boiler stays, in any case, are but substitutes for real strength of construction. They would be of no service applied to a sphere subject to internal pressure; and the power of resistance would be exactly that of the metal to sustain the strain exerted upon all its parts alike. The manner in which stays are frequently employed renders them a source of weakness rather than an element of strength. When

the strain is direct the power of resistance of the stay is equal to the weight it would sustain without tearing it asunder; but when the position of the stay is oblique to the point of resistance, any calculation of their theoretic strength or value is attended with certain difficulties. All boilers should be sufficiently stayed to insure safety, and the material of which they are made, their shape, strength, number, location and mode of attachment to the boiler, should all be duly and intelligently considered. Boiler stays should never be subjected to a strain of more than one-eighth of their breaking strength. The strength of boiler stays may be calculated by multiplying the area in inches between the stays by the pressure in pounds per square inch.

Rule for finding the strain allowed on a diagonal boiler head brace or stay; also rule for finding the number of stays required for a certain size crown sheet.

Iron stays should not be subjected to a greater stress than from 7,000 to 9,000 pounds per square inch of section, and if they are located obliquely, the diameter will need to be increased an amount that depends on the angle of the stay to the shell. Find the area in square inches to be supported by the stay, and multiply it by the pressure per square inch, multiply the product by the length of the diagonal stay, and divide the result by the perpendicular length from the flat surface to the end of the stay. The quotient will be the stress on the stay, and to obtain the diameter, divide the stress by the allowable stress per square inch of section, and the quotient by .7854. The square root of the last quotient will be the diameter of the stay.

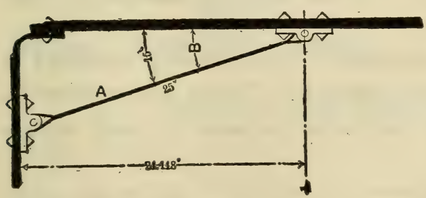


Fig. 261. Diagonal boiler stay.

Thus, in the accompanying diagram, we wish to find the diameter of the diagonal stay *A*, which supports an area 6″ x 8″ or 48 square inches. The

length of the stay is 25″, and the perpendicular distance between the stayed surface and the end of the stay is 24.148″. The boiler pressure is 100 pounds gauge, so that the pressure on the surface supported will be 48 x 100 or 4,800 pounds. We multiply 4,800 by 25 and divide the product by 24.148″, which gives 4,970, nearly. The quotient of 4,970, divided by 7,000 equals .71; .71, divided by .7854 equals .9039, and the square root of this is .95 or .95″, the diameter of a stay that will support 48 square inches in the position shown.

A convenient formula for finding the diameter of oblique stays is,

$$D \text{ equals } 1.13\sqrt{\frac{A\,P}{L \cos B}}$$

D equals diameter of the stay.
A " area in square inches to be supported.
P " pressure per square inch.
L " safe load per square inch of stay section.
B " angle between the shell and the stay.

Using the preceding problem as an example and referring to the same diagram, we have angle B equal to 15°, and all the other dimensions as previously given. Therefore,

$$D \text{ equals } 1.13\sqrt{\frac{48 \times 100}{7000 \times .96593}}$$

The diameter of the stay, when the above is simplified, is .9526″, or practically 1″. A rule for finding the pitch of stays for any flat surface is given below.

1. A safe formula for the strength of stayed flat surfaces is that given by Unwin's Machine Design. When the spacing of the stays is desired, assuming that it is the same in each direction, we have,

$$a \text{ equals } 3\,t\sqrt{\frac{f}{2\,p}}$$

where a equals spacing of stays or rivets in inches, f equals safe working strength of the plate, t equals thickness of plate, and p equals boiler pressure. Expressed as a rule, this reads: Divide the safe strength of the plate by twice the pressure; extract the square root of the quotient and multiply the final result by three times the thickness of the plate. The result will be the spacing of the stays in inches. For example, boiler pressure 100 pounds, plate 1/2 inch thick, safe strength of plate, 10,000 pounds per square inch; $2p$ equals 2 x 100 equals 200; $f/2p$ equals 10000/200 equals 50; $\sqrt{50}$ equals 7.07; $3t$ equals 3/2 equals 1-1/2 equals 1.5; 7.07 x 1.5 equals 10.6 for the spacing. In making such a calculation care must be exercised not to assume too high values for the strength of the plate. It is not safe to count on more than 60,000 pounds for the strength of steel plates and 40,000 for iron. The working strength must be taken not higher than 1/6 of this, or 10,000 for steel and 6,666 for iron, and lower values still would be better, say 9,000 for steel and 6,000 for iron.

2. **The safe** pressure for a boiler to carry, so far as the flat, stayed surfaces are concerned, may be found from the above formula by transposing it a little, as follows: —

$$p \text{ equals } \frac{9\ t^2 f}{2\ a^2}$$

Now, applying this to the above example, we have p equals $\frac{9 \times .5^2 \times 10000}{2 \times 110.25}$ which equals $\frac{9 \times .25 \times 10000}{2 \times 110.25}$ and which after reduction equals $\frac{22500}{220.50}$ equals 102, or substantially the pressure assumed in the first example.

RIVETED AND LAP-WELDED FLUES.

The following table shall include all riveted and lap-welded flues exceeding 6 inches in diameter and not exceeding 40 inches in diameter not otherwise provided by law, as required by U. S. Gov.

TABLE OF STEAM PRESSURE PER SQUARE INCH ALLOWABLE ON RIVETED AND LAP-WELDED FLUES MADE IN SECTIONS.

Thickness of material required.	Greatest length of sections allowable, 5 feet.				Greatest length of sections allowable, 3 feet.									
	Least thickness of material allowable.				Least thickness of material allowable.									
	.18 inch.	.20 inch.	.21 inch.	.21 inch.	.22 inch.	.22 inch.	.23 inch	.24 inch.	.25 inch.	.26 inch.	.27 inch.	.28 inch.	.29 inch.	.30 inch.
	Diameter of flues.													
	Over 6 and not over 7 inches.	Over 7 and not over 8 inches.	Over 8 and not over 9 inches.	Over 9 and not over 10 inches.	Over 10 and not over 11 inches.	Over 11 and not over 12 inches.	Over 12 and not over 13 inches.	Over 13 and not over 14 inches.	Over 14 and not over 15 inches.	Over 15 and not over 16 inches.	Over 16 and not over 17 inches.	Over 17 and not over 18 inches.	Over 18 and not over 19 inches.	Over 19 and not over 20 inches.
	Pounds pres-sure.	*Pounds* pres-sure.	*Pounds* pres-sure.	*Pounds* pres-sure.	*Pounds* pres-sure.	*Pounds* pres-sure.	*Pounds* pres-sure.	*Pounds* pres-sure.	*Pounds* pres-sure.	*Pounds* pres-sure.	*Pounds* pres-sure.	*Pounds* pres-sure.	*Pounds* pres-sure.	*Pounds* pres-sure.
.18-inch..	189	....	....	....	....	....	....	....	....	....	....	....	....	....
.19-inch..	194	....	....	....	....	....	....	....	....	....	....	....	....	....
.20-inch..	199	184	....	....	....	....	....	....	....	....	....	....	....	....
.21-inch..	204	189	179	174	....	....	....	....	....	....	....	....	....	....
.22-inch..	....	194	184	179	172	158	....	....	....	....	....	....	....	....
.23-inch..	....	199	189	184	179	165	152	....	....	....	....	....	....	....
.24-inch..	....	204	194	189	187	172	158	147	....	....	....	....	....	....
.25-inch..	....	....	199	194	195	179	165	153	143	....	....	....	....	....
.26-inch..	....	....	204	199	203	186	172	159	148	139	....	....	....	....
.27-inch..	....	....	....	204	211	193	178	165	154	143	136	....	....	....
.28-inch..	....	....	....	....	....	200	185	172	160	150	141	134	....	....
.29-inch..	....	....	....	....	....	207	191	178	166	155	146	138	131	....
.30-inch..	....	....	....	....	....	....	198	184	172	161	152	143	135	129
.31-inch..	....	....	....	....	....	....	205	190	178	166	156	148	140	133
.32-inch..	....	....	....	....	....	....	....	196	183	171	161	153	145	137
.33-inch..	....	....	....	....	....	....	....	202	189	177	167	157	149	141
.34-inch..	....	....	....	....	....	....	....	....	194	182	172	162	154	146
.35-inch..	....	....	....	....	....	....	....	....	200	187	177	167	159	150
.36-inch..	....	....	....	....	....	....	....	....	....	193	182	172	163	154
.37-inch..	....	....	....	....	....	....	....	....	....	....	187	177	168	159
.38-inch..	....	....	....	....	....	....	....	....	....	....	....	182	172	163
.39-inch..	....	....	....	....	....	....	....	....	....	....	....	....	176	167
.40-inch..	....	....	....	....	....	....	....	....	....	....	....	....	....	172

TABLE OF STEAM PRESSURE PER SQUARE INCH ALLOWABLE ON RIVETED AND LAP-WELDED FLUES MADE IN SECTIONS.—Continued.

Thickness of material required.	Greatest length of sections allowable, 3 ft.			Greatest length of sections allowable, 30 inches.										
	Least thickness of material allowable.			Least thickness of material allowable.										
	.31 inch.	.32 inch.	.33 inch.	.34 inch.	.35 inch.	.36 inch.	.37 inch.	.38 inch.	.39 inch.	.40 inch.	.41 inch.	.42 inch.	.43 inch.	.44 inch.
	Diameter of flues.													
	Over 20 and not over 21 inches.	Over 21 and not over 22 inches.	Over 22 and not over 23 inches.	Over 23 and not over 24 inches.	Over 24 and not over 25 inches.	Over 25 and not over 26 inches.	Over 26 and not over 27 inches.	Over 27 and not over 28 inches.	Over 28 and not over 29 inches.	Over 29 and not over 30 inches.	Over 30 and not over 31 inches.	Over 31 and not over 32 inches.	Over 32 and not over 33 inches.	Over 33 and not over 34 inches.
	Pounds pressure.	*Pounds* pressure.	*Pounds* pressure.	*Pounds* pressure.	*Pounds* pressure.	*Pounds* pressure.	*Pounds* pressure.	*Pounds* pressure.	*Pounds* pressure.	*Pounds* pressure.	*Pounds* pressure.	*Pounds* pressure.	*Pounds* pressure.	*Pounds* pressure.
.31-inch..	126	......	......	......	......	......	......	......	......	......	......	......	......	......
.32-inch..	131	125	......	......	......	......	......	......	......	......	......	......	......	......
.33-inch..	135	129	122	......	......	......	......	......	......	......	......	......	......	......
.34-inch..	139	132	126	......	......	......	......	......	......	......	......	......	......	......
.35-inch..	143	137	130	......	......	......	......	......	......	......	......	......	......	......
.36-inch..	147	141	134	......	......	......	......	......	......	......	......	......	......	......
.37-inch..	151	145	138	......	......	......	......	......	......	......	......	......	......	......
.38-inch..	155	149	142	......	......	......	......	......	......	......	......	......	......	......
.39-inch..	159	152	146	......	......	......	......	......	......	......	......	......	......	......
.40 inch..	163	156	150	......	......	......	......	......	......	......	......	......	......	......
.41-inch..	167	160	153	......	......	......	......	......	......	......	......	......	......	......
.42-inch..	......	164	157	......	......	......	......	......	......	......	......	......	......	......
.43 inch..	......	......	160	......	......	......	......	......	......	......	......	......	......	......
.34-inch..	......	......	......	121	......	......	......	......	......	......	......	......	......	......
.35-inch..	......	......	......	125	120	......	......	......	......	......	......	......	......	......
.36-inch..	......	......	......	129	123	119	......	......	......	......	......	......	......	......
.37-inch..	......	......	......	132	127	122	117	......	......	......	......	......	......	......
.38-inch..	......	......	......	136	130	125	121	116	......	......	......	......	......	......
.39-inch .	......	......	......	139	133	129	124	119	115	......	......	......	......	......
.40-inch..	......	......	......	143	137	132	127	123	118	115	......	......	......	......
.41-inch..	......	......	......	147	141	135	130	126	121	118	114	......	......	......
.42-inch..	......	......	......	151	144	139	133	129	124	120	117	112	......	......
.43-inch..	......	......	......	154	147	142	137	132	127	123	119	115	112	......
.44-inch..	......	......	......	157	151	145	140	135	130	126	122	118	115	110
.45-inch..	......	......	......	......	155	148	143	138	133	129	124	121	117	113
.46-inch..	......	......	......	......	......	151	146	141	136	132	127	124	120	116

For any flue requiring more pressure than is given in table, the same will be determined by proportion of thickness to any given pressure in table to thickness for pressure required, as per example: A flue not over 19 inches diameter and 3 feet long, requires a thickness of .39 of an inch for 176 pounds pressure; what thickness would be required for 250 lbs. pounds pressure?

176 : .39 : : 250 : .5539,

or a thickness of .554 inch.

Or, if .39 inch thickness gives a pressure of 176 lbs., what will .554 inch thickness give?

.39 : 176 : : .554 : 250 pounds required.

And all such flues shall be made in sections, according to their respective diameters, not to exceed the lengths prescribed in the table and such sections shall be properly fitted one into the other and substantially riveted, and the thickness of material required for any such flue of any given diameter shall in no case be less than the least thickness prescribed in the table for any such given diameter; and all such flues may be allowed the prescribed working steam pressure, if in the opinion of the inspectors, it is deemed safe to make such allowance. And inspectors are therefore required, from actual measurement of each flue, to make such reduction from the prescribed working steam pressure for any material deviation in the uniformity of the thickness of the material, or for any material deviation in the form of the flue from that of a true circle, as in their judgment the safety of the boiler may require.

Riveted and lap-welded flues of any thickness of material, diameter, and length of sections prescribed in the table, may be made in sections of any desired length, exceeding the maximum length allowed by the table, by reducing the prescribed pressure

in proportion to the increased length of section, according to the following rule: —

Rule. — Multiply the pressure in the table allowed for any prescribed thickness of material and diameter of flue by the greatest length, in feet, of sections allowable for such flue, and divide the product by the desired length of sections, in feet, from center line to center line of rivets, in the circular seams of such sections, and the quotient will give the working steam pressure allowable.

Example. — Taking a flue in the table 24 inches in diameter, required to be made in sections not exceeding 2.5 feet in length, and having a thickness of material of .44 of an inch, and allowed a pressure of 157 lbs., and it is desired to make this flue in sections 5 feet in length.

Then we have

$$\frac{157 \times 2.5}{5} = 78.5 \text{ lbs. pressure allowable.}$$

THICKNESS OF MATERIAL REQUIRED FOR TUBES AND FLUES NOT OTHERWISE PROVIDED FOR.

Tubes and flues not exceeding 6 inches in diameter, and made of any required length; and

Lap-welded flues required to carry a working steam pressure not to exceed 60 lbs. per square inch, and having a diameter not exceding 16 inches, and a length not exceeding 18 feet; and

Lap-welded flues required to carry a steam pressure exceeding 60 lbs. per square inch, and not exceeding 120 lbs. per square inch, and having a diameter not exceeding 16 inches and a length not exceeding 18 feet, and made in sections not exceeding 5 feet in length, and fitted properly one into the other, and substantially riveted; and

All such flues shall have a thickness of material according to their respective diameters, as prescribed in the following table: —

Outside diameter.	Thickness.	Outside diameter.	Thickness.	Outside diameter.	Thickness.
Inches.	*Inch.*	*Inches.*	*Inch.*	*Inches.*	*Inch.*
1	.072	$3\frac{1}{4}$	.120	9	.180
$1\frac{1}{4}$	.072	$3\frac{1}{2}$	.120	10	.203
$1\frac{1}{2}$	.083	$3\frac{3}{4}$	.120	11	.220
$1\frac{3}{4}$	.095	4	.134	12	.229
2	.095	$4\frac{1}{2}$	.134	13	.238
$2\frac{1}{4}$	.095	5	.148	14	.248
$2\frac{1}{2}$	.109	6	.165	15	.259
$2\frac{3}{4}$	.109	7	.165	16	.270
3	.109	8	.165		

Tubes, water pipes and steams pipes, made of steel manufactured by the Bessemer process, may be used in any steam boiler when the material from which pipes are made does not contain more than .06 per cent of phosphorus and .04 per cent of sulphur, to be determined by analysis by the manufacturers, verified by them, and copy furnished the user for each order tested; which analysis shall, if deemed expedient by a properly qualified person, be verified by an outside test at the expense of the manufacturer of the tubes or pipes. No tube increased in thickness by welding one tube inside of another, shall be allowed for use.

Seamless copper or brass tubes, not exceeding three-fourths of an inch in diameter, may be used in the construction of water tube pipe boilers or generators, when liquid fuel is used. There may also be used in their construction copper or brass steam drums, not exceeding 14 inches in diameter, of a thickness of material not less than five-eighths of an inch, and copper or brass steam drums 12 inches in diameter and under, having a thickness of material not less than one-half inch. All the tubes and drums referred to in this paragraph shall be made from ingots or blanks drawn down to size without a seam. Water-tube boilers or gen-

erators so constructed may be used for power purposes with none other than liquid fuel.

Lap-welded flues not exceeding 6 inches in diameter may be made of any required length without being made in sections. And all such lap-welded flues and riveted flues not exceeding 6 inches in diameter may be allowed a working steam pressure not to exceed 225 lbs. per square inch, if deemed safe by the inspectors.

Lap-welded flues exceeding 6 inches in diameter and not exceeding 16 inches in diameter, and not exceeding 18 feet in length, and required to carry a steam pressure not exceeding 60 lbs. per square inch, shall not be required to be made in sections.

Lap-welded and riveted flues exceeding 6 inches in diameter and not exceeding 16 inches in diameter, and not exceeding 18 feet in length, and required to carry a steam pressure exceeding 60 lbs. per square inch, and not exceeding 120 lbs. per square inch, may be allowed, if made in sections not exceeding 5 feet in length and properly fitted one into the other, and substantially riveted.

On all boilers built for high pressures, a bronze or brass-seated stop-cock or valve shall be attached to the boiler between all check valves and all steam and feed pipes and boilers, in order to facilitate access to connections. Where such cocks or valves exceed $1\frac{1}{2}$ inches in diameter, they must be flanged to boiler. The stop-valves attached to main steam-pipes may, however, be made of cast-iron or other suitable material. The date referred to above applies to this paragraph only.

All copper steam pipes shall be flanged to a depth of not less than four times the thickness of the material in the pipes, and all such flanging shall be made to a radius not to exceed the thickness of the material in such pipes. And all such pipes shall have a thickness of material according to the working steam pressure

allowed, and such thickness of material shall be determined by the following rule: —

Rule. — Multiply the working steam pressure in pounds per square inch allowed the boiler by the diameter of the pipe in inches, then divide the product by the constant whole number 8000, and add .0625 to the quotient; the sum will give the thickness of the material required.

Example. — Let 175 lbs. = working steam pressure per square inch allowed the boiler,

5 inches = diameter of the pipe,

8000 = a constant.

Then we have: —

$\frac{175 \times 5}{8000} + .0625 = .1718 +$ thickness of material in decimals of an inch.

The flanges of all copper steam pipes over three inches in diameter shall be made of bronze or brass composition, and shall have a thickness of material of not less than four times the thickness of material in the pipes plus .25 of an inch; and all such flanges shall have a boss of sufficient thickness of material projecting from the back of the flange a distance of not less than three times the thickness of material in the pipe; and all such flanges shall be counter-bored in the face to fit the flange of the pipe; and the joints of all copper steam pipes shall be made with a sufficient number of good and substantial bolts to make such joints at least equal in strength to all other parts of the pipe.

The terminal and intermediate joints of all wrought iron and homogeneous steel feed and steam pipes over 2 inches in diameter and not over 5 inches in diameter, other than on pipe or coil boilers or steam generators, shall be made of wrought iron, homogeneous steel, or malleable iron flanges, or equivalent material; and all such flanges shall have a depth through the bore of not

less than that equal to one-half of the diameter of the pipe to which any such flange may be attached; and such bores shall taper slightly outwardly toward the face of the flanges; and the ends of such pipes shall be enlarged to fit the bore of the flanges, and they shall be substantially beaded into a recess in the face of each flange. But where such pipes are made of extra heavy lap-welded steam pipe, the flanges may be attached with screw threads; and all joints in bends may be made with good and substantial malleable iron elbows, or equivalent material.

All feed and steam pipes not over 2 inches in diameter may be attached at their terminals and intermediate joints with screw threads by flanges, sleeves, elbows, or union couplings; but where the ends of such pipes at their terminal joints are screwed into material in the boiler, drum or other connection having a thickness of not less than $\frac{1}{2}$ inch, the flanges of such terminal joints may be dispensed with. Where any such pipes are not over one inch in diameter and any of the terminal ends are to be attached to material in the boiler or connection having a thickness of less than $\frac{1}{2}$ inch, a nipple shall be firmly screwed into the boiler or connection against a shoulder, and such pipe shall be screwed firmly into such nipple. And should inspectors deem it necessary for safety, they may require a jam nut to be screwed onto the inner end of any such nipple.

The word "terminal" shall be interpreted to mean the points where steam or feed pipes are attached to such appliances on boilers, generators or engine, as are placed on such to receive them.

All lap-welded iron or steel steam pipes over 5 inches in diameter, or riveted wrought iron or steel steam pipes over 5 inches in diameter, in addition to being expanded into tapered holes and substantially beaded into recess in face of flanges, as provided in preceding paragraph for steam and feed pipes exceeding 2 inches and not exceeding 5 inches in diameter, shall be substantially and

firmly riveted, with good and substantial rivets, through the hubs of such flanges; and no such hubs shall project from such flanges less than 2 inches in any case.

Steam pipes of iron or steel, when lap-welded by hand or machine, with their flanges welded on, shall be tested to a hydrostatic pressure of at least double the working pressure of the steam to be carried and properly annealed after all the work requiring fire is finished. When an affidavit of the manufacturer is furnished that such test has been made and annealed, they may be used for power purposes.

WROUGHT IRON WELDED PIPE.

DIMENSIONS, WEIGHTS, ETC., OF STANDARD SIZES FOR STEAM, GAS, WATER, OIL, ETC.

1 inch and below are butt-welded, and tested to 300 pounds per square inch hydraulic pressure.

$1\frac{1}{4}$ inch and above are lap-welded, and tested to 500 pounds per square inch hydraulic pressure.

Inside Diameter.	Outside Diameter.	External Circumference.	Length of Pipe per sq. ft. of outside surface.	Internal Area.	External Area.	Length of Pipe containing one cubic foot.	Weight per ft. of length.	No. of threads per inch of screw.	Contents in *Gallons per foot.	Weight of Water per foot of Length.
Inch.	Inches.	Inches.	Feet.	Inches.	Inches.	Feet.	Lbs.			Lbs.
$\frac{1}{8}$	.40	1.272	9.44	.012	.129	2500.	.24	27	.0006	.005
$\frac{1}{4}$	.54	1.696	7.075	049	229	1385.	.42	18	.0026	.021
$\frac{3}{8}$	.67	2.121	5.657	.110	.358	751.5	.56	18	.0057	.047
$\frac{1}{2}$	.84	2.652	4.502	.196	.554	472.4	.84	14	.0102	.085
$\frac{3}{4}$	1.05	3.299	3.637	.441	.866	270.	1.12	14	.0230	.190
1	1 31	4.134	2.903	.785	1.357	166.9	1.67	$11\frac{1}{2}$	.0408	.349
$1\frac{1}{4}$	1.66	5.215	2.301	1.227	2.164	96.25	2.25	$11\frac{1}{2}$	.0638	.527
$1\frac{1}{2}$	1.9	5.969	2.01	1.767	2.835	70.65	2.69	$11\frac{1}{2}$	.0918	.760
2	2.37	7.461	1.611	3.141	4.430	42.36	3.66	$11\frac{1}{2}$	.1632	1.356
$2\frac{1}{2}$	2.87	9.032	1.328	4.908	6.491	30.11	5.77	8	.2550	2.116
3	3.5	10.996	1 091	7.068	9.621	19.49	7.54	8	.3673	3.049
$3\frac{1}{2}$	4.	12.566	.955	9.621	12.566	14.56	9.05	8	.4998	4.155
4	4.5	14.137	.849	12.566	15.904	11.31	10.72	8	.6528	5.405
$4\frac{1}{2}$	5.	15.708	.765	15.904	19.635	9.03	12.49	8	.8263	6.851
5	5.56	17.475	.629	19.635	24.299	7.20	14.56	8	1.020	8.500
6	6.62	20.813	.577	28.274	34.471	4.98	18.76	8	1.469	12.312
7	7.62	23.954	.505	38.484	45.663	3.72	23.41	8	1.999	16.662
8	8.62	27.096	.444	50.265	58.426	2.88	28.34	8	2.611	21.750
9	9 68	30.433	.394	63.617	73.715	2. 26	34.67	8	3.300	27.500
10	10.75	33.772	.355	78.540	90.792	1.80	40.64	8	4.081	34.000

PULSATION IN STEAM-BOILERS.

Pulsation in steam-boilers, though not discernible to the eye, as in animated nature, goes on intermittently in some boilers whenever they are in use. It is induced by weakness and want of capacity in the boiler to supply the necessary quantity of steam, and sometimes is caused by the boiler being badly designed, thereby admitting of a great disproportion between the heating-surface and steam-room. Boilers are frequently found in factories that were originally not more than of sufficient capacity to furnish the necessary quantity of steam, but, as business increased, it became necessary to increase the pressure and also the speed of the engine; or, perhaps to replace it with a larger one, which has to be supplied with steam from the same boiler. The result is, each time the valve opens to admit steam to the cylinder, about one-third of the whole quantity in the boiler is admitted, thus lowering the pressure; the next instant, under the influence of hard firing, or, perhaps, a forced draught, the steam is brought to the former pressure, and so on; this lessening and increasing the pressure continues while the engine is in motion, which has an effect on the boiler similar to the breathing of an animal.

The strains induced by this pulsation are transmitted to the weakest places, viz., the line of the rivet holes, and that marked by the tool in the process of calking; the result is, the plate is broken in two, as shown in the above cut. The manner in which the break takes place may be illustrated by filing a small nick, or drilling a small hole, in a piece of hoop or band-iron, and then bending back

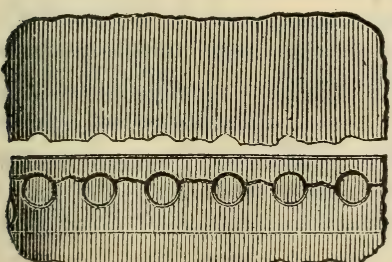


Fig. 262. Cracked plate.

and forth, when it will be discovered that the material will break just at that point, however slight the nick or small the hole may be. Pulsation is frequently very severe in the boilers of tug-boats when commencing to start a heavy tow, and also in locomotives when starting long trains. Some frightful explosions of the boilers of tug-boats and locomotives have occurred under such circumstances. Pulsation, if permitted to continue, is sure to effect the destruction of the boiler. It is always made manifest by the vibrations of the pointers on steam gauges, or an unsteadiness in the mercury column. It may be remedied, to a certain extent, by adding a larger steam dome, but this has a tendency to weaken the boiler and render it more unsafe. The only sure preventive of such a silent and destructive agent is to have the boiler of sufficient capacity in the first place.

WEIGHT OF SQUARE AND ROUND IRON PER LINEAR FOOT.

SIDE OR DIAM.	Weight, Square.	Weight, Round.	SIDE OR DIAM.	Weight, Square.	Weight. Round.	SIDE OR DIAM.	Weight, Square.	Weight, Round.
1/16	.013	.01	2	13.52	10.616	5	84.48	66.35
1/8	.053	.041	1/8	15.263	11.988	1/4	93.168	73.172
3/16	.118	.093	1/4	17.112	13.44	1/2	102.24	80.304
1/4	.211	.165	3/8	19.066	14.975	3/4	111.756	87.776
3/8	.475	.373	1/2	21.12	16.588			
1/2	.845	.663	5/8	23.292	18.293	6	121.664	95.552
5/8	1.32	1.043	3/4	25.56	20.076	1/4	132.04	103.704
3/4	1.901	1.493	7/8	27.939	21.944	1/2	142.816	112.16
7/8	2.588	2.032				3/4	154.012	120.96
			3	30.416	23.888			
1	3.38	2.654	1/4	35.704	28.04	7	165.632	130.048
1/8	4.278	3.359	1/2	41.408	32.515	1/4	177.672	139.544
1/4	5.28	4.147	3/4	47.534	37.332	1/2	190.136	149.328
3/8	6.39	5.019				3/4	203.024	159.456
1/2	7.604	5.972	4	54.084	42.464			
5/8	8.926	7.01	1/4	61.055	47.952	8	216.336	169.856
3/4	10.352	8.128	1/2	68.448	53.76			
7/8	11.883	9.333	3/4	76.264	59.9	9	273.792	215.04

WATER COLUMNS.

Every boiler should be equipped with a safety water column. Next to keeping the steam pressure within the limits of safety, the most important point to be observed in operating steam boilers is the maintenance of the proper water level. If the water level is too low, there is danger of burning the tubes and plates and, perhaps, of wrecking the boiler; if it is too high, water is liable to be carried along with the steam and cause damage in the engine, while a constant variation in the water level produces a waste of fuel and unsteady pressure, and impairs the life of the boiler. Safety water columns have been devised for the purpose of insuring owners of steam boilers against accidents of this kind. They are so arranged that any variation in the water level beyond reasonable limits will be loudly proclaimed by means of a suitable steam whistle.

STEAM-GAUGES.

The object of the steam-gauge is to indicate the steam pressure in the boiler, in order that it may not be increased far above that at which the boiler was originally considered safe; and it is as a provision against this contingency that a really good gauge is a necessity where steam is employed, for no guide at all is vastly better than a false one. The most essential requisites of a good steam-gauge are, that it be accurately graduated, and that the material and workmanship be such that no sensible deterioration may take place in the course of its ordinary use. The pecuniary loss arising from any considerable fluctuation of the pressure of steam has never been properly considered by the proprietors of engines. If steam be carried too high, the surplus will escape through the safety-valve, and all the fuel consumed to produce such excess is so much dead loss. On the other hand, if there be at any time too little steam, the engine will run too slow, and every lathe, loom, or other machine driven by it, will lose its speed and, of course, its effective power in the same pro-

portion. A loss of one revolution in ten at once reduces the productive power of every machine driven by the engine ten per cent, and loses to the proprietor ten per cent of the time of every workman employed to manage such machine. In short, the loss of one revolution in ten diminishes the productive capacity of the whole concern ten per cent, so long as such reduced rate continues; while the expenses of conducting the shop (rent, wages, insurance, etc.) all run on as if everything was in full motion. A variation to this amount is a matter of frequent occurrence, and is, indeed, unavoidable, unless the engineer is afforded facilities to prevent it. A very little reflection will satisfy any one that it must be a very small concern, indeed, in which a half-hour's continuance of it would not produce a result more than enough to defray the cost of a very expensive instrument to prevent it. If the engineer, to avoid this loss, keeps a surplus of steam constantly on hand, he is constantly wasting the steam, and consequently, fuel, thus incurring another loss, which, though less alarming than the first, will yet be serious and render any instrument most desirable which can prevent it. It is, therefore, of great importance to the proprietors of engines to have an instrument which can constantly indicate the pressure in the steam-boilers with accuracy. This would enable the engineer to keep his steam at a constant pressure, thus avoiding waste of fuel on the one hand, and the still more serious loss of the productive power of the shop on the other. An instrument, therefore, constantly indicating the pressure of steam, reliable in its character, and, with ordinary care, not subject to derangement, is evidently a desideratum both to the engineer and proprietor. The importance of such an instrument, as a preventive of explosion, and of the frightful consequences to life and limb and ruinous pecuniary results of such disaster, is obvious on the slightest consideration; but the value of the instrument, in the economical results of its daily use, is by no means properly appreciated.

SAFETY-VALVES.

The form and construction of this indispensable adjunct to the steam boiler are of the highest importance, not only for the preservation of life and property, which would, in the absence of that means of "safety" be constantly jeopardized, but also to secure the durability of the steam boiler itself. And yet, judging from the manner in which many things called safety-valves have been constructed of late years, it would appear that the true principle by which *safety* is sought to be secured by this most valuable adjunct is either not well understood, or is disregarded by many engineers and boiler makers.

Boiler explosions have in many cases occurred when, to all appearances, the safety-valves attached have been in good working order; and coroners' juries have not unfrequently been puzzled, and sometimes guided to erroneous verdicts by scientific evidence adduced before them, tending to show that nothing was wrong with the safety-valves, and that the devastating catastrophies could not have resulted from overpressure, because in such case the safety-valve would have prevented them. It is supposed that a gradually increasing pressure can never take place if the safety-valve is rightly proportioned and in good working order. Upon this assumption, universally acquiesced in, when there is no accountable cause, explosions are attributed to the "sticking" of the valves, or to "bent" valve-stems, or inoperative valve-springs. As the safety-valve is the sole reliance, in case of neglect or inattention on the part of the engineer or fireman, it is important to examine its mode of working closely. Safety-valves are usually provided with a spindle or guide-pin, attached to the under side, and passing through a cross-bar within the boiler, directly under the seating of the valve, which may be seen in

the cut below. Now, it is evident that if this guide-pin becomes bent from careless handling, the safety-valve may be rendered almost inoperative, and, instead of releasing the pressure at the point indicated, it will turn sideways, and allow only a small aperture for the escape of steam, and, further, it will not return perfectly to its seat; hence, a leaky valve is the result, and to overcome this difficulty, ignorant engineers and firemen generally resort to extra weighting; and it is not uncommon to find double or treble the weight

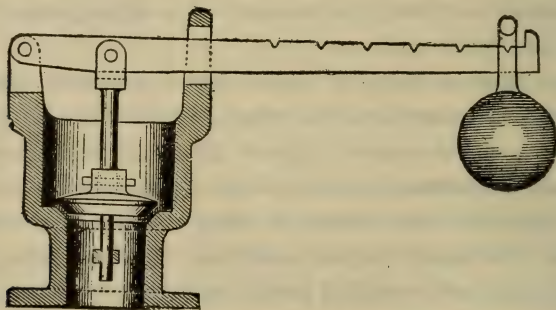


Fig. 263. Lever and weight safety-valve.

corresponding to the pressure required in the boiler. Another difficulty is that the safety-valve levers sometimes get bent, and the weight, consequently, hangs on one side of the true center; this, it will be seen, causes the valve to rest more heavily on one side than on the other, and the greater the added weight the greater the difficulty. The seats of safety-valves should be examined frequently to see that no corrosion has commenced; as valves, especially if leaky, become corroded and often *stick fast*, so that no little force is required to raise them. If, when a safety-valve is properly weighted, it should be found leaking, do not put on extra weights, but immediately make an examination, and in all probability the seat or guide-pin will be found corroded, or there will be foreign matter between the valve and its

seat. By taking the lever in the hand and raising it from its seat a few times, any substance that may have kept it from its seat will be dislodged; or it may turn out on examination that the lever had deviated from some cause from a true center. Such difficulties can be easily righted, but extra weight should never be added, as it only aggravates the trouble instead of remedying it. When the weight of the safety-valve is set on the lever at safe working pressure, or at the distance from the fulcrum necessary to maintain the pressure required to work the engine, any extra length of lever should then be cut off as a precaution, to prevent the moving out of the weight on the lever, for the purpose of increasing the pressure, as, while the lever remains sufficiently long, the weight can be increased to a dangerous extent without attracting any attention; while if the lever is cut off at the point at which the safe working pressure is designated, any extra increase of pressure can only be accomplished by adding more weight to the lever, which is tolerably sure to attract the attention of some one interested in the preservation of the lives and property of persons in the immediate vicinity.

The bolts that form the connection between the lever, fulcrum and valve-stem should be made of brass, in order to prevent the possibility of corrosion, "sticking" or becoming magnetized, as it is termed; and for the same reason, the valve and seat should be made of two different metals. When safety-valves become leaky they should be taken out and reground on their seats, for which purpose pulverized glass, flour of emery, or the fine grit or mud from grinding stone troughs are the most suitable material; but whether they leak or not, they should be taken apart at least once a year and all the working parts cleaned, oiled and readjusted. The safety-valve is designed on the assumption that it will rise from its seat under the statical pressure in the boiler, when this pressure exceeds the exterior pressure on the valve, and that it will remain off its seat sufficiently far to permit all the

steam which the boiler can produce to escape around the edges of the valve. The problem then to be solved is: What amount of opening is necessary for the free escape of the steam from the boiler under a given pressure? The area of a safety-valve is generally determined from formulae based on the velocity of the flow of steam under different pressures, or upon the results of experiments made to ascertain the area necessary for the escape of all the steam a boiler could produce under a given pressure. But as the fact is now generally recognized by engineers that valves do not rise appreciably from their seats under varying pressures, it is of importance that in practice the outlets round their edges should be greater than those based on theoretical considerations. The next point to be considered is how high any safety valve will rise under the influence of a given pressure. This question cannot be determined theoretically, but has been settled conclusively by Burg, of Vienna, who made careful experiments to determine the actual rise of safety-valves above their seats. His experiments show that the rise of the valve diminishes rapidly as the pressure increases.

TABLE SHOWING THE RISE OF SAFETY-VALVES, IN PARTS OF AN INCH, AT DIFFERENT PRESSURES.

Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
12	20	35	45	50	60	70	80	90
$\frac{1}{36}$	$\frac{1}{48}$	$\frac{1}{54}$	$\frac{1}{65}$	$\frac{1}{86}$	$\frac{1}{86}$	$\frac{1}{132}$	$\frac{1}{168}$	$\frac{1}{168}$

Taking ordinary safety-valves, the average rise for pressures from 10 to 40 pounds is about $\frac{1}{40}$ of an inch, from 40 to 70 pounds about $\frac{1}{80}$, and from 70 to 90 pounds about $\frac{1}{120}$ of an inch. The following table gives the result of a series of experiments made at the Novelty Iron Works, New York, for the purpose of determining the exact area of opening necessary for

safety-valves, for each square foot of heating surface, at different boiler pressures.

TABLE.

Pressure in Boiler in pounds above the atmosphere.	Area of Orifice in Sq. In. for each Sq. Ft. of Heating Surface.	Pressure in Boiler in pounds above the atmosphere.	Area of Orifice in Sq. In. for each Sq. Ft. of Heating Surface.	Pressure in Boiler in pounds above the atmosphere.	Area of Orifice in Sq. In. for each Sq. Ft. of Heating Surface.
0.25	.022794	10	.005698	70	.001015
0.5	.021164	20	.003221	80	.000892
1	.018515	30	.002244	90	.000796
2	.014814	40	.001723	100	.000719
3	.012345	50	.001398	150	.000481
4	.010582	60	.001176	200	.000364
5	.009259				

TABLE OF COMPARISON BETWEEN EXPERIMENTAL RESULTS AND THEORETICAL FORMULAE.

Boiler Pressure, 45 pounds.			Boiler Pressure, 75 pounds.		
Heating Surface.	Area of opening found by experiment.	Area of opening according to formulae.	Heating Surface.	Area of opening found by experiment.	Area of opening according to formulae.
Sq. Ft.	Sq. Ins.	Sq. Ins.	Sq. Ft.	Sq. Ins.	Sq. Ins.
100	.089	.09	100	.12	.12
200	.180	.19	200	.24	.24
500	.45	.48	500	.59	.59
1000	.89	.94	1000	1.20	1.18
2000	1.78	1.90	2000	2.40	2.37
5000	4.46	4.75	5000	6.00	5.95

Now, if we compare the area of openings, according to these experiments, with Zeuner's formula, which is entirely theoretical, it will be observed that the results from the two sources are almost identical, or so nearly so as not to make any material difference. In the absence of any generally recognized rule, it is customary for engineers and boiler-makers to proportion safety-valves according to the heating surface, grate surface, or horse-power of the boiler. While one allows one inch of area of safety-valve to 66 square feet of heating surface, another gives one inch area of safety-valve to every four horse power; while a third proportions his by the grate surface — it being the custom in such cases to allow one inch area of safety-valves to 2 square feet of grate surface. This latter proportion has been proved by long experience and a great number of accurate experiments, to be capable of admitting of a free escape of steam without allowing any material increase of the pressure beyond that for which the valve is loaded, even when the fuel is of the best quality, and the consumption as high as 24 pounds of coal per hour per square foot of grate surface, providing, of course, that all the parts are in good working order. It is obvious, however, that no valve can act without a slight increase of pressure, as, in order to lift at all, the internal pressure must exceed the pressure due to the load.

The lift of safety-valves, like all other puppet valves, decreases as the pressure increases; but this seeming irregularity is but what might be required of an orifice to satisfy conditions in the flow of fluids, and may be explained as follows: A cubic foot of water generated into steam at one pound pressure per square inch above the atmosphere, will have a volume of about 1,600 cubic feet. Steam at this pressure will flow into the atmosphere with a velocity of 482 feet per second. Now, suppose the steam was generated in five minutes, or in 300 seconds, and the area of an orifice to permit its escape as fast as it is generated be re-

quired, 1600 x 144 ÷ (482 x 300) will give the area of the orifice, $1\frac{3}{5}$ square inches. If the same quantity of water be generated into steam at a pressure of 50 pounds above the atmosphere, it will possess a volume of 440 cubic feet and will flow into the atmosphere with a velocity of 1791 feet per second. The area of an orifice, to allow this steam to escape in the same time as in the first case, may be found as follows: 440 x 144 ÷ (1791 x 300), the result will be $\frac{3}{25}$ square inches, or nearly $\frac{1}{8}$ of a square inch, the area required. It is evident from this that a much less lift of the same valve will suffice to discharge the same weight of steam under a high pressure than under a low one, because the steam under a high pressure not only possesses a reduced volume, but a greatly increased velocity; it is also obvious from these considerations, that a safety-valve, to discharge steam as fast as the boiler can generate it, should be proportioned for the lowest pressure.

RULES.

Rule. — For finding the weight necessary to put on a safety-valve lever when the area of valve, pressure, etc., are known: Multiply the area of valve by the pressure in pounds per square inch; multiply this product by the distance of the valve from the fulcrum; multiply the weight of the lever by one-half its length (or its center of gravity); then multiply the weight of valve and stem by their distance from the fulcrum; add these last two products together, subtract their sum from the first product, and divide the remainder by the length of the lever; the quotient will be the weight required.

EXAMPLE.

Area of valve, 12 in.	65	13	8
Pressure, 65 lbs.	12	16	4
Fulcrum, 4 in.	780	208	32

Length of lever, 32 in.	4	13
Weight of lever, 13 lbs.		
	———	———
Weight of valve and stem, 8 lbs.	3120	208
	240	32
	———	———
	32)2880	240
	———	
	90 lbs.	

Rule for finding the pressure per square inch when the area of valve, weight of ball, etc., are known: Multiply the weight of ball by length of lever, and multiply the weight of lever by one-half its length (or its center of gravity); then multiply the weight of valve and stem by their distance from the fulcrum. Add these three products together. This sum, divided by the product of the area of valve, and its distance from the fulcrum, will give the pressure in pounds per square inch.

EXAMPLE.

Area of valve, 7 in.	50	12	6
Fulcrum, 3 in.	30	15	3
	———	——	——
Length of lever, 30 in.	1500	180	18
Weight of lever, 12 lbs.	180		
		———	
Weight of ball, 50 lbs.	18		7
Weight of valve and stem, 6 lbs.			
	———		
	21)1698		3
	———		——
	80.85 lbs.		21

Rule for finding the pressure at which a safety-valve is weighted when the length of the lever, weight of ball, etc., are known: Multiply the length of lever in inches by the weight of ball in pounds; then multiply the area of valve by its distance

from the fulcrum; divide the former product by the latter; the quotient will be the pressure in pounds per square inch.

EXAMPLE.

Length of lever, 24 in.	52	7
Weight of ball, 52 lbs.	24	3
Fulcrum, 3 in.	208	21
Area of valve, 7 in.	104	
	21)1248	
	59.42 lbs.	

The above rule, though very simple, cannot be said to be exactly correct, as it does not take into account the weight of the lever, valve and stem.

Rule for finding center of gravity of taper levers for safety-valves: Divide the length of lever by two (2); then divide the length of lever by six (6), and multiply the latter quotient by width of large end of lever less the width of small end, divided by width of large end of lever plus the width of small end. Subtract this product from the first quotient, and the remainder will be the distance in inches of the center of gravity from large end of lever.

EXAMPLE.

Length of lever	36 in.
Width of lever at large end	3 "
Width of lever at small end	2 "

36 divided by 2 = 18 minus 1.2 = 16.8 in. 36 divided by 6 = $6 \times 1 = 6$ divided by 5 = 1.2.

Center of gravity from large end, 16.8 in.

The safety-valve has not received that attention from engineers and inventors which its importance as a means of safety

so imperatively deserves. In the construction of most other kinds of machinery, continual efforts have been made to secure and insure accuracy; while in the case of the safety-valve, very little improvement has been made either in design or fitting. It is difficult to see why this should be so, when it is known that deviations from exactness, though trifling in themselves, when multiplied, not only affect the free action and reliability of machines, but frequently result in serious injury, more particularly in the case of safety-valves.

Safety-valves should never be made with rigid stems, as, in consequence of the frequent inaccuracy of the other parts, the valve is prevented from seating, thereby causing leakage; as a remedy for which, through ignorance or want of skill, more weight is added on the lever, which has a tendency to bend the stem, thus rendering the valve a source of danger instead of a means of safety. The stem should, in all cases, be fitted to the valve with a ball and socket joint, or a tapering stem in a straight hole, which will admit of sufficient vibration to accommodate the valve to its seat. It is also advisable that the seats of safety-valves, or the parts that bear, should be as narrow as circumstances will permit, as the narrower the seat the less liable the valve is to leak, and the easier it is to repair when it becomes leaky.

All compound or complicated safety-valves should be avoided, as a safety-valve is, in a certain sense, like a clock — any complication of its parts has a tendency to affect its reliability and impair its accuracy.

It has been too much the custom heretofore for owners of steam boilers to disregard the advice and suggestions of their own engineers and firemen, even though men of intelligence and experience, and to be governed entirely by the advice of self-styled experts and visionary theorists.

TABLE OF HEATING SURFACE IN SQUARE FEET IN HORIZONTAL TUBULAR BOILERS.

Diam. of Boiler in inches	24	30	32	34	36	38	40	42	44	48
⅔ Heating surface of shell per foot of length.	4.19	5.24	5.57	5.93	6.28	6.63	6.98	7.73	7.68	8.38
Diameter of Tube or Flue in inches.	2	2½	3	3½	4	4½	5	6	7	8
Whole External Heating surface per foot length.	.524	.655	.785	.916	1.05	1.18	1.31	1.57	1.83	2.09

Diam. of Boiler in inches	50	52	54	56	58	60	62	64	66	68	70	72
⅔ Heating surface of shell per foot of length.	8.73	9.08	9.42	9.77	10.12	10.47	10.82	11.17	11.52	11.87	12.22	12.57
Diameter of Tube or Flue in inches.	9	10	11	12	13	14	15	16	17	18	19	20
Whole External Heating surface per foot length.	2.36	2.62	2.88	3.14	3.40	3.66	3.93	4.19	4.45	4.71	4.96	5.24

CENTRIFUGAL FORCE.

The centrifugal force of a body depends upon its weight W in pounds; distance R in feet it is from the center of rotation, and the number of revolutions N it makes about that center each minute and equals $\frac{W\,R\,N^2}{2933}$.

Multiply the weight in pounds by radius in feet, by square of number of revolutions, and divide by 2933 = centrifugal force in pounds.

CHAPTER XVIII.

THE WATER TUBE BOILER.

The water tube boiler has been a growth of many years and of many different minds. There are some two and a half million horse-power in daily service in the United States alone, and the number is rapidly increasing. Large orders for this type of boiler have often been repeated, adding proof that its principles are correct and appreciated by those having them in use and in charge. This being the case, purchasers should note well the points of difference in the various water tube boilers claiming their attention, and particularly see that the claims made for them are embodied in their actual construction. The general principles of construction and operation of this class of steam boilers are now well known to engineers and steam users. In selecting a water tube boiler there are several vital points to be considered: —

1st. Straight and smooth passages through the headers of ample area, insuring rapid and uninterrupted circulation of the water.

2d. The baffling of the gases (without throttling or impeding the circulation of the water) in such a way that they are compelled to pass over every portion of the heating surface.

3d. Sufficient liberating surface in the steam drums to insure dry steam, with large body of water in reserve to draw from.

4th. A steam reservoir or steam drum.

5th. Simplicity in construction; accessibility for cleaning and inspection.

6th. A header, which in its design provides for the unequal expansion and contraction.

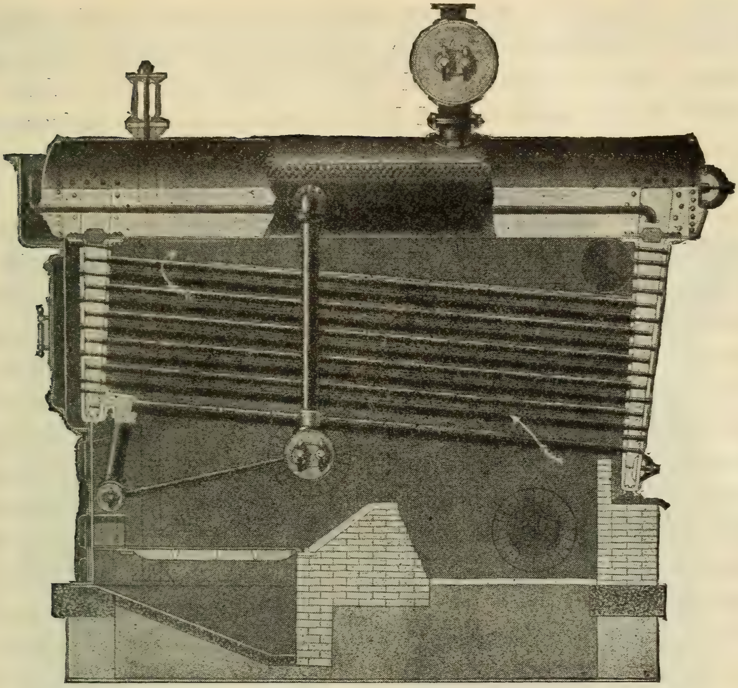


Fig. 264. O'Brien horizontal safety water tube boiler.
Manufactured by the John O'Brien Boiler Works Company, of St. Louis, U. S. A.

This type of water tube boiler when provided with a cross-drum to reduce the head room required is adapted to, and is oftentimes used in heating plants.

Down draft furnace. — A great many of these boilers are fitted with the down draft furnaces, and the above illustration shows the style of same, together with the manner in which they are connected.

A full and complete description of these furnaces is given on page 522.

Description. — In construction, this type of boiler consists

simply of a front and rear water leg or header, made approximately rectangular in shape, overhead combination steam and water drum or drums and with circulating water tubes, as shown in cut, which extend between and connect both front and rear headers, being thoroughly expanded into the tube sheets. The tubes are inclined on a pitch of one inch to the foot and the rear header being longer than the front one, the overhead drum connecting both headers lies perfectly level when the boiler is set in position. The connection of the headers with the combined steam and water drum is made in such a manner as to give practically the same area as the total area of the tubes, so there is no contraction of area in the course of circulation; and extending between and connecting the inside faces of the water legs, which form end connections between these tubes and the combined steam and water drums or shells, placed above and parallel with them, also a steam drum above these, assures absolutely dry steam and a large steam space, also a large water space. The water legs are made larger at the top, about 11 inches wide, and at the bottom about 7 inches wide, which is a great advantage, allowing the globules of steam to pass quickly up the water legs to the steam and water drums. The water, as it sweeps along the drums, frees itself of steam; then it goes down the back connection until it meets the inclined tubes, meeting on its passage a gradually increasing temperature, till the furnace is again reached, where the steam formed on the way is directly carried up in the drum as before. The tubes extend between and connect both the front and rear headers and are thoroughly expanded into the tube sheets. Opposite the end of each tube there is an oval hand-hole slightly larger than the tube proper through which it can be withdrawn. It will be noted that the throat of each water leg is $1\frac{1}{3}$ times the total tube area. The rapid and unimpeded circulation tends to keep the inside surface clean and floats the scale-making sediment along until it reaches the back

water leg, where it is carried down and settles in the bottom of leg, where it is blown off at regular intervals.

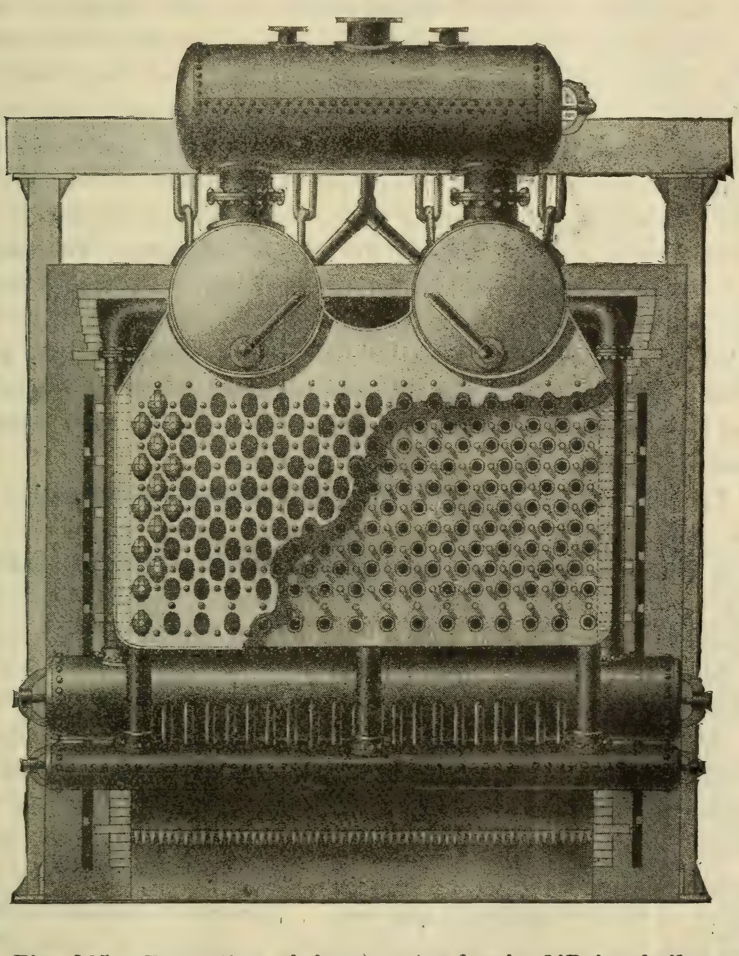


Fig. 265. Formation of front water leg in O'Brien boiler.

Steadiness of water level.—The large area of surface at water line and the ample passages for circulation, secure a steadiness

of water level peculiar to this type. This is a most important point in boiler construction and should always be considered when comparing boilers. The water legs are stayed by hollow stay-bolts of hydraulic tubing of large diameter, so placed that two stays support each tube and hand-hole and are subjected to only very slight strain. Being made of heavy material, they form the strongest parts of the boiler and its natural supports. The water legs are joined to the shell by flanged and riveted joints and the drum is cut away at these two points to make connection with inside of water leg, the opening thus made being strengthened by special stays, so as to preserve the original strength. The shells are cylinders with heads dished to form part of a true sphere. The sphere is everywhere as strong as the circular seam of the cylinder, which is well known to be twice as strong as the side seam; therefore, the heads require no stays. Both the cylinder and the spherical heads are, therefore, free to follow their natural lines of expansion when put under pressure.

The illustration on page 505 plainly shows the formation of the front water leg or header in this type of water tube boiler.

It will be seen that the hand plates are all oval in shape, allowing each one to be removed from its respective hole; also, the manner of bracing with hollow stay-bolts is shown.

Note that the feed pipes for supplying boiler run back to rear water leg and discharge therein.

Walling in. — In setting the boiler, its front water leg is placed firmly on a set of strong, cast-iron columns bolted and braced together by the door frames and dead-plates and forming the fire front. This is the fixed end. The rear water legs rest on rollers which are free to move on cast-iron plates firmly set in the masonry of the low and solid rear wall. Thus the boiler and its walls are each free to move separately during expansion or contraction, without loosening any joints in the masonry.

On the lower, and between the upper tubes, are placed light

fire-brick tiles. The lower tier extends from the front water leg to within a few feet of the rear one, leaving there an upward passage across the rear ends of the tubes for the flame. The upper tier closes into the rear water leg and extends forward to within a few feet of the front one, thus leaving an opening for the gases in front. The side tiles extend from side walls to tile bars and close up to the front water leg and front wall, and leave open the final uptake for the waste gases.

The gases being thoroughly mingled in their passage between the staggered tubes, the combustion is more complete, and the gases impinging against the heating surface perpendicularly, instead of gliding along the same longitudinally, the absorption of the gas is more thorough. The draft area, being much larger than in fire tube boilers, gives ample time for the absorption of the heat of the gases before their exit to the chimney.

DESCRIPTION OF THE HEINE SAFETY BOILER.

The boiler is composed of lap-welded wrought-iron tubes extending between and connecting the inside faces of two "water legs," which form the end connections between these tubes and a combined steam and water drum or "shell" placed above and parallel with them. Boilers over 200 horse-power have two such shells. These end chambers are of approximately rectangular shape, drawn in at top to fit the curvature of the shells. Each is composed of a *head plate* and a *tube sheet* flanged all around and joined at bottom and sides by a butt strap of same material, strongly riveted to both. The water legs are further stayed by hollow stay-bolts of hydraulic tubing of large diameter, so placed that two stays support each tube and hand-hole and are subjected to only very slight strain. Being made of heavy metal, they form the strongest parts of the boiler and its natural supports. The

water legs are joined to the shell by flanged and riveted joints, and the drum is cut away at these two points to make connection

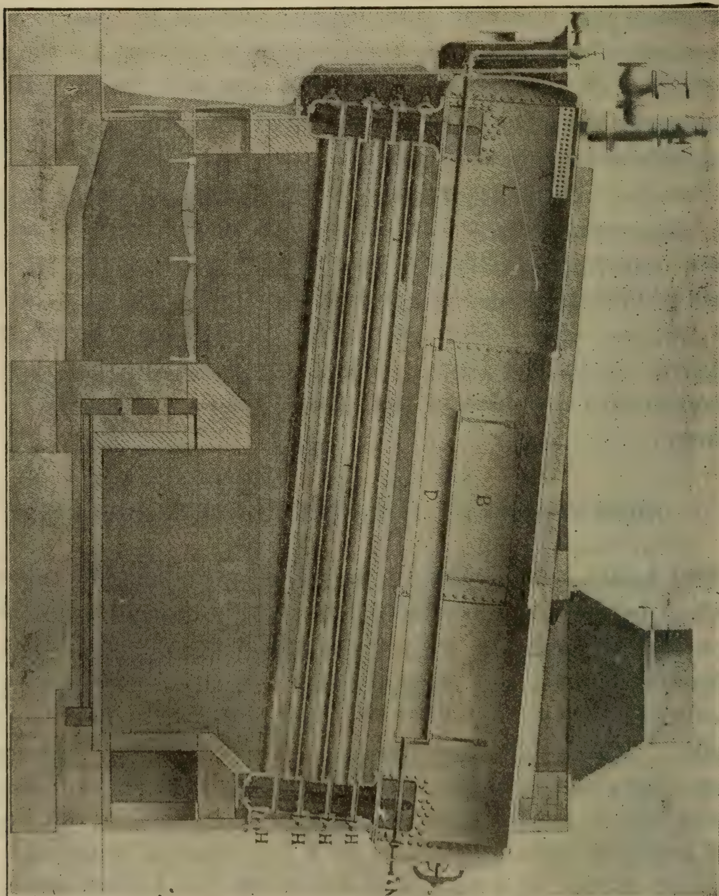


Fig. 266. Sectional view of the Heine safety boiler.

with inside of water leg, the opening thus made being strengthened by bridges and special stays so as to preserve the original strength.

The shells are cylinders with heads dished to form parts of a true sphere. The sphere is everywhere as strong as the circle seam of the cylinder, which is well known to be twice as strong as its side seam. Therefore, these heads require no stays. Both the cylinder and its spherical heads are, therefore, free to follow their natural lines of expansion when put under pressure. Where flat heads have to be braced to the sides of the shell, both suffer local distortions where the feet of the braces are riveted to them, making the calculations of their strength fallacious. This they avoid entirely by their dished heads. To the bottom of the front head a flange is riveted, into which the feed-pipe is screwed. This pipe is shown in the cut with angle valve and check valve attached. On top of shell, near the front end, is riveted a steam nozzle or saddle, to which is bolted a tee. This tee carries the steam valve on its branch, which is made to look either to front, rear, right or left; on its top the safety valve is placed. The saddle has an area equal to that of stop valve and safety valve combined. The rear head carries a blow-off flange of about same size as the feed flange, and a manhead curved to fit the head, the manhole supported by a strengthening ring outside. On each side of the shell a square bar, the tile-bar, rests loosely in flat hooks riveted to the shell. This bar supports the side tiles, whose other ends rest on the side walls, thus closing the furnace or flue on top. The top of the tile-bar is two inches below low water line. The bars rise from front to rear at the rate of one inch in twelve. When the boiler is set, they must be exactly level, the whole boiler being then on an incline, i. e., with a fall of one inch in twelve from front to rear. It will be noted that this makes the height of the steam space in front about two-thirds the diameter of the shell, while at the rear the water occupies two-thirds of the shell, the whole contents of the drum being equally divided between steam and water. The importance of this will be explained hereafter.

The tubes extend through the tube sheets, into which they are expanded with roller expanders; opposite the end of each and in the head plates, is placed a hand-hole of slightly larger diam-

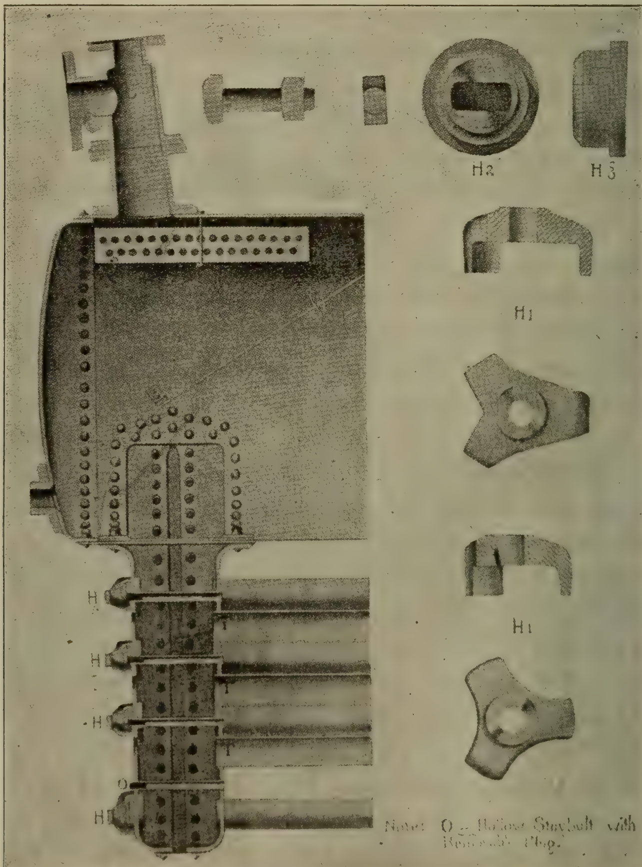


Fig. 267. Details of construction — Heine boiler.

eter than the tube, and through which it can be withdrawn. These hand-holes are closed by small cast-iron hand-hole plates, which, by an ingenious device for locking, can be removed in a

few seconds to inspect or clean a tube. The accompanying cut shows these hand-hole plates marked *H*. In the upper corner one is shown in detail, H^2 being the top view, H^3 the side view of the plate itself, the shoulder showing the place for the gasket. H^1 is the yoke or crab placed outside to support the bolt and nut. Inside of the shell is located the mud drum *D*, placed well below the water line, usually parallel to and 3 inches above the bottom of the shell. It is thus completely immersed in the hottest water in the boiler. It is of oval section, slightly smaller than the manhole, made of strong sheet-iron with cast-iron heads. It is entirely inclosed except about 18 inches of its upper portion at the forward end, which is cut away nearly parallel to the water line. Its action will be explained below. The feed-pipe *F* enters it through a loose joint in front; the blow-off pipe *N* is screwed tightly into its rear-head, and passes by a steam-tight joint through the rear-head of the shell. Just under the steam nozzle is placed a dry pan or dry pipe *A*. A deflection plate *L* extends from the front head of the shell, inclined upwards, to some distance beyond the mouth or throat of the front water leg. It will be noted that the throat of each water leg is large enough to be the practical equivalent of the total tube area, and that just where it joins the shell it increases gradually in width by double the radius of the flange.

Erection and walling in. — In setting the boiler, its front water leg is placed firmly on a set of strong cast-iron columns, bolted and braced together by the door frames, deadplate, etc., and forming the fire front. This is the fixed end. The rear water leg rests on rollers, which are free to move on cast-iron plates firmly set in the masonry of the low and solid rear wall. Wherever the brickwork closes in to the boiler, broad joints are left which are filled in with tow or waste saturated with fireclay, or other refractory but pliable material. Thus the boiler and its walls are each free to move separately during expansion or con-

traction without loosening any joints in the masonry. On the lower, and between the upper tubes, are placed light firebrick tiles. The lower tier extends from the front water leg to within a few feet of the rear one, leaving there an upward passage across the rear ends of the tubes for the flame, etc. The upper tier closes in to the rear water leg and extends forward to within a few feet of the front one, thus leaving the opening for the gases in front. The side tiles extend from side walls to tile bars and close up to the front water leg and front wall, and leave open the final uptake for the waste gases over the back part of the shell, which is here covered above water line with a rowlock of firebrick resting on the tile bars. The rear wall of the setting and one parallel to it arched over the shell a few feet forward, form the uptakes. On these and the rear portion of the side walls is placed a light sheet-iron hood, from which the breeching leads to the chimney. When an iron stack is used, this hood is stiffened by *L* and *T* irons so that it becomes a truss carrying the weight of such stack and distributing it to the side walls.

Heine boiler and its operation. — The boiler being filled to middle water line, the fire is started on the grate. The flame and gases pass over the bridge wall and under the lower tier of tiling, finding in the ample combustion chamber space, temperature and air supply for complete combustion, before bringing the heat in contact with the main body of the tubes. Then, when at its best, it rises through the spaces between the rear ends of the tubes, between rear water leg and back end of the tiling, and is allowed to expand itself on the entire tube heading surface without meeting any obstruction. Ample space makes leisurely progress for the flames, which meet in turn all the tubes, lap round them, and finally reach the second uptake at the forward end of the top tier of tiling, with their temperature reduced to less than 900° Fahrenheit. This has been measured here, while wrought iron would melt just above the lower tubes at

rear end, showing a reduction of temperature of over 1,800° Fahr. between the two points. As the space is studded with water tubes, swept clean by a positive and rapid circulation, the absorption of this great amount of heat is explained. The gases next travel under the bottom and sides of shell and reach the uptake at just the proper temperature to produce the draft required. This varies, of course, according to chimney, fuel required, etc. With boilers running at their rated capacity, 450° Fahrenheit are

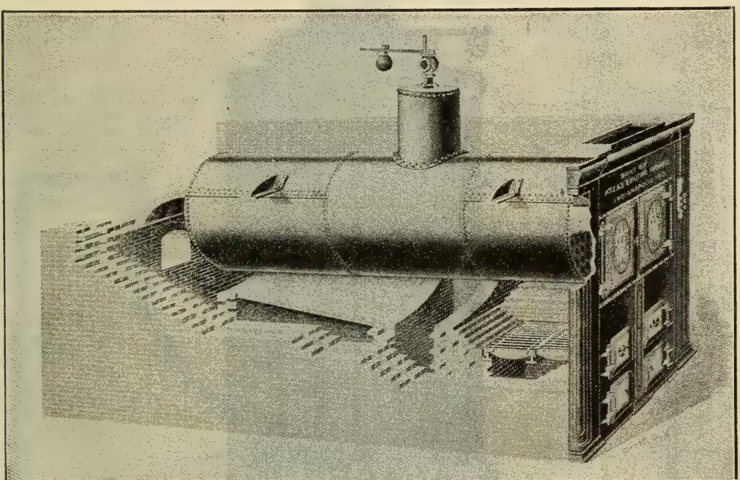


Fig. 268. A furnace that is used in the East.

seldom exceeded. Meanwhile, as soon as the heat strikes the tubes, the circulation of the water begins. The water nearest the surface of the tubes becoming warmer, rises, and as the tubes are higher in front, this water flows towards the front water leg where it rises into the shell, while colder water from the shell falls down the rear water leg to replace that flowing forward and upward through the tubes. This circulation, at first slow, in-

creases in speed as soon as steam begins to form. Then the speed with which the mingled current of steam and water rises in the forward water leg will depend on the difference in weight of this mixture, and the solid and slightly colder water falling down the rear water leg. The cause of its motion is exactly the same as that which produces draft in a chimney.

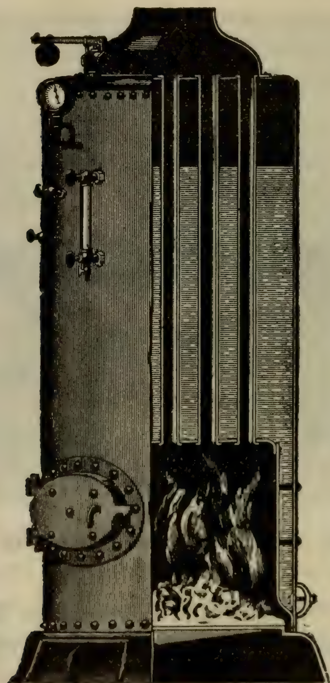


Fig. 269. Plain vertical tubular boiler.

This cut shows the place for gauge cocks and water glass in an upright boiler.

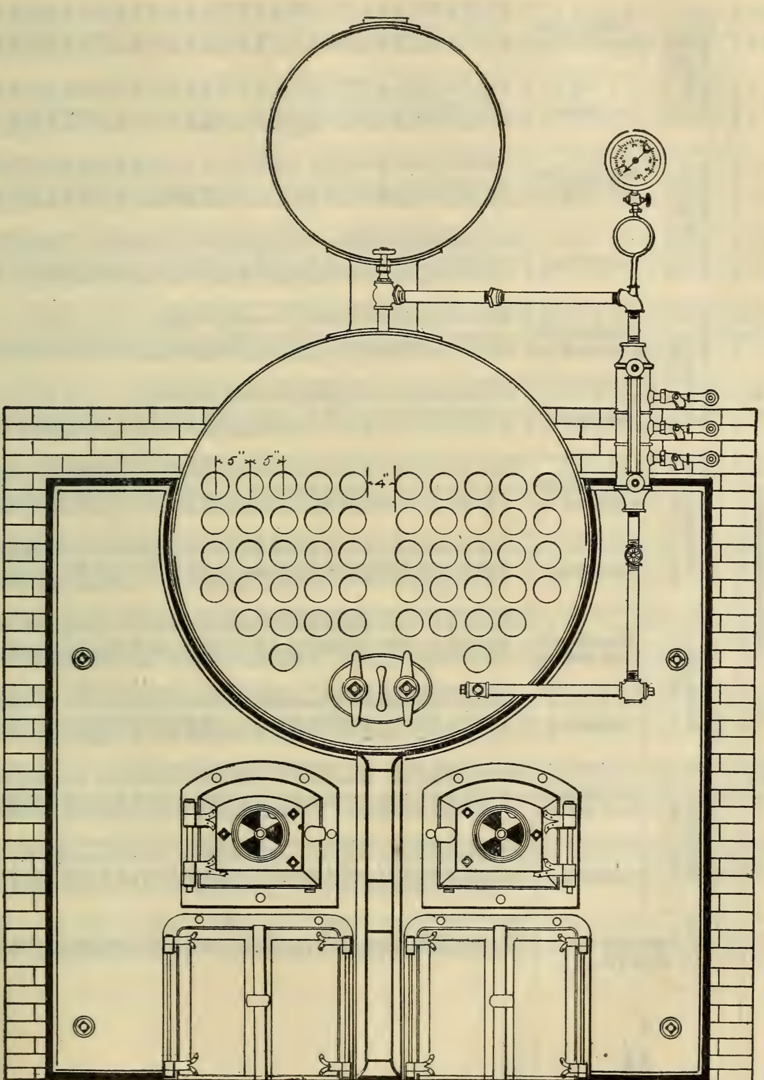


Fig. 270. Showing the water-column in its proper place.

Table of Pressures allowable on Boilers made since February 28, 1872, by the United States Government.

Diameter of boiler.	Thickness of plates.	45,000 tensile strength, 1-6, 7,500.		50,000 tensile strength, 1-6, 8,333.3.		55,000 tensile strength, 1-6, 9,166.6.		60,000 tensile strength, 1-6, 10,000.		65,000 tensile strength, 1-6, 10,833.3.		70,000 tensile strength, 1-6, 11,666.6.	
		Pressure.	20 per cent additional.	Pressure.	20 per cent additional	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.	Pressure.	20 per cent additional.
36 inches........	.1875	78.12	93.74	86.8	104.16	95.48	114.57	104.16	124.99	112.84	135.4	121.52	145.82
	.21	87.5	105	97.21	116.65	106.94	128.33	116.66	139.99	126.38	151.65	136.11	163.33
	.23	95.83	114.99	106.47	127.76	117.12	140.54	127.77	153.32	138.41	166.09	149.07	178.88
	.25	104.16	124.99	115.74	138.88	127.31	152.77	138.88	166.65	150.46	180.55	162.03	194.43
	.26	108.33	129.99	120.37	144.44	132.4	158.88	144.44	173.32	156.48	187.77	168.51	202.21
	.29	120.83	144.99	134.25	161.11	147.68	177.21	161.11	193.33	174.53	209.43	187.96	225.55
	.3125	130.2	156.24	144.67	173.6	159.14	190.96	173.6	208.32	188.07	225.68	202.5	243.04
	.33	137.5	165	152.77	183.32	168.05	201.66	183.33	219.99	198.61	238.33	213.88	256.65
	.35	145.83	174.99	162.03	194.43	178.23	213.87	194.44	233.32	210.64	252.76	226.84	272.20
	.375	156.25	187.5	173.61	208.33	190.97	229.16	208.33	249.99	225.69	270.83	243.05	291.66
38 inches........	.1875	74.01	88.81	82.23	98.67	90.46	108.54	98.68	118.41	106.9	128.28	115.13	138.16
	.21	82.89	99.46	92.1	110.52	101.31	121.57	110.52	132.62	119.73	143.67	128.93	154.71
	.23	90.78	108.93	100.87	121.04	110.96	133.15	121.05	145.26	131.13	157.35	141.22	169.46
	.25	98.68	118.41	109.64	131.56	120.61	144.73	131.57	157.88	142.54	171.04	153.5	184.20
	.26	102.63	123.15	114.03	136.83	125.43	150.51	136.84	164.2	148.24	177.88	159.64	191.56
	.29	114.47	137.36	127.19	152.62	139.91	167.89	152.63	183.15	165.35	198.42	178.06	213.67
	.3125	123.35	148.02	137.06	164.46	150.76	180.91	164.47	197.36	178.17	213.8	191.88	230.25
	.33	130.26	156.31	144.73	173.67	159.2	191.04	173.68	208.41	188.15	225.78	202.62	243.14
	.35	138.15	165.78	153.5	184.21	168.85	202.62	184.21	221.05	199.56	239.47	214.91	257.89
	.375	148	177.60	164.73	197.67	180.91	217.09	197.36	236.83	213.81	256.57	230.26	276.31
40 inches........	.1875	70.31	84.37	78.12	93.74	85.93	103.11	93.75	112.5	101.56	121.87	109.37	131.24
	.21	78.75	94.50	87.49	104.98	96.24	115.48	105	126	113.74	136.48	122.49	146.98
	.23	86.25	103.5	95.83	114.99	105.41	126.49	115	138	124.58	149.49	134.16	160.99
	.25	93.75	112.5	104.16	124.99	114.58	137.49	125	150	135.41	162.49	145.83	174.99
	.26	97.5	117	108.33	129.99	119.16	142.99	130	156	140.83	168.99	151.66	181.99
	.29	108.75	130.5	120.83	144.99	132.91	159.49	145	174	157.08	188.49	169.16	202.99
	.3125	117.18	140.61	130.2	156.24	143.22	171.86	156.25	187.45	169.27	203.12	182.29	218.74
	.33	123.75	148.5	137.49	164.98	151.24	181.48	165	198	178.74	214.48	192.49	230.98
	.35	131.25	157.5	145.83	174.99	160.41	192.49	175	210	189.58	227.49	204.16	244.99
	.375	140.62	168.74	156.24	187.48	171.87	206.24	187.5	225	203.12	243.74	218.74	262.48

42 inches........	.1875	66.96	80.35	74.40	89.28	81.84	98.20	89.28	107.13	96.72	116.06	104.16	124.99
	.21	75	90	83.32	99.99	91.66	109.99	100	120	108.33	129.99	116.66	139.99
	.23	82.14	98.56	91.23	109.51	100.39	120.46	109.52	131.42	118.65	142.38	127.77	153.32
	.25	89.28	107.13	99.2	119.04	109.12	130.94	119.04	142.84	128.96	154.75	138.88	166.65
	.26	92.85	111.42	103.17	123.8	113.49	136.18	123.8	148.56	134.12	160.94	144.44	173.32
	.29	103.57	124.28	115.07	138.08	126.57	151.85	138.09	165.7	149.6	179.52	161.11	193.33
	.3125	111.6	133.92	124	148.8	136.4	163.68	148.74	178.56	161.2	193.44	173.61	208.23
	.33	117.85	141.42	130.94	157.12	144.04	172.84	157.14	188.56	170.23	204.27	183.33	219.99
	.35	125	150	138.88	166.65	152.77	183.32	166.66	199.99	180.55	216.66	194.44	233.32
	.375	133.92	160.7	148.8	178.56	163.68	196.40	178.57	214.28	193.45	232.14	208.33	249.99
44 inches........	.1875	63.92	76.7	71.02	85.22	78.12	93.74	85.22	102.26	92.32	110.78	99.42	119.3
	.21	71.59	85.9	79.54	95.44	87.49	104.98	95.45	114.54	103.4	124.08	111.36	133.63
	.23	78.4	94.08	87.12	104.54	95.83	114.99	104.54	125.44	113.25	135.9	121.96	146.35
	.25	85.22	102.26	94.69	113.62	104.16	124.99	113.63	136.35	123.1	147.72	132.56	159.07
	.26	88.63	106.35	98.48	118.17	108.33	129.99	118.18	141.81	128.02	153.62	137.87	165.44
	.29	98.86	118.63	109.84	131.80	120.83	144.99	131.81	158.17	142.79	171.33	153.78	184.53
	.3125	106.53	127.83	118.36	142.03	130.2	156.24	142.04	170.44	153.88	184.65	165.71	198.85
	.33	112.5	135	124.99	149.98	137.49	164.98	150	180	162.49	194.98	174.99	209.98
	.35	119.31	143.17	132.57	159.08	145.83	174.99	159.09	190.9	172.34	206.8	185.6	222.72
	.375	127.81	153.37	142.04	170.44	156.24	187.48	170.45	204.54	184.65	221.58	198.86	238.63
46 inches........	.1875	61.14	73.36	67.93	81.51	74.72	89.66	81.51	97.81	88.31	105.97	95.1	114.12
	.21	68.47	82.16	76.08	91.29	83.69	100.42	91.3	109.56	98.91	118.69	106.52	127.82
	.23	75	90	83.33	100	91.66	109.99	100	120	108.33	129.99	116.66	139.99
	.25	81.52	97.82	90.57	108.68	99.63	119.55	108.69	130.42	117.75	141.3	126.8	152.16
	.26	84.78	101.73	94.2	113.04	103.62	124.34	113.44	135.64	122.46	146.95	131.88	158.25
	.29	94.56	113.47	105.07	126	115.57	138.68	126.09	151.3	136.59	163.92	147.1	176.52
	.3125	101.9	122.28	113.21	135.86	124.54	149.44	135.86	163.03	147.19	176.62	158.51	190.21
	.33	107.6	129.12	119.56	143.47	131.52	157.82	143.97	172.16	155.43	186.51	167.39	200.86
	.35	144.13	136.95	126.8	152.16	139.49	167.38	152.17	182.6	164.85	197.82	177.53	213.03
	.375	122.28	146.73	135.86	163.03	149.45	179.34	163.04	195.64	176.62	211.94	190.21	228.25
48 inches........	.1875	58.59	70.30	65.1	78.12	71.61	85.93	78.12	93.74	84.63	101.55	91.13	109.35
	.21	65.62	78.74	72.91	87.49	80.2	96.24	87.49	104.98	94.79	113.74	102.08	122.49
	.23	71.87	86.24	79.85	95.82	87.84	105.4	95.83	114.99	103.81	124.57	111.8	133.16
	.25	78.12	93.74	86.8	104.16	95.48	114.57	104.16	124.99	112.84	135.4	121.52	145.82
	.26	81.25	97.50	90.27	108.32	99.3	119.16	108.33	129.99	117.36	140.83	126.38	151.65
	.29	90.62	108.74	100.69	120.82	110.76	132.91	120.83	144.99	130.9	157.08	140.97	169.16
	.3125	97.65	117.18	108.5	130.2	119.35	143.22	130.21	156.25	141.05	169.26	151.9	182.28
	.33	103.12	123.74	114.58	137.49	126.04	151.24	137.5	165	148.95	178.74	160.41	192.49
	.35	109.37	131.24	121.52	145.83	133.67	160.4	145.83	174.99	157.98	189.57	170.13	204.15
	.375	117.18	140.61	130.2	156.24	143.22	171.86	156.25	187.50	169.27	203.12	182.29	218.74
54 inches........	.1875	52.08	62.49	57.87	69.44	63.65	76.38	69.44	82.44	75.23	90.27	81.01	97.21
	.21	58.33	69.99	64.81	77.77	71.29	85.54	77.77	93.32	84.25	101.1	90.74	108.88
	.23	63.88	76.65	70.98	85.17	78.08	93.69	85.18	102.21	92.28	110.73	99.38	119.25
	.25	69.44	83.32	77.16	92.59	84.87	101.84	92.59	111.10	100.3	120.36	108.02	129.62
	.26	72.22	86.66	80.24	96.28	88.27	105.92	96.29	115.54	104.31	125.17	112.44	134.8
	.29	80.55	96.66	89.5	107.40	98.45	118.14	107.41	128.88	116.35	139.62	125.3	150.36

Table of Pressures allowable on Boilers made since February 28, 1872, by the United States Government. — *Continued.*

Diameter of boiler.	Thickness of plates.	45,000 tensile strength, 1-6, 7,500. Pressure.	45,000: 20 per cent additional.	50,000 tensile strength, 1-6, 8,333.3. Pressure.	50,000: 20 per cent additional.	55,000 tensile strength, 1-6, 9,166.6. Pressure.	55,000: 20 per cent additional.	60,000 tensile strength, 1-6, 10,000. Pressure.	60,000: 20 per cent additional.	65,000 tensile strength, 1-6, 10,833.3. Pressure.	65,000: 20 per cent additional.	70,000 tensile strength, 1-6, 11,666.6. Pressure.	70,000: 20 per cent additional.
54 inches.......	.3125	86.8	104.16	96.44	115.73	106.09	127.30	115.55	138.66	125.38	150.45	135.03	162.03
	.33	91.66	109.99	101.84	122.22	112.03	134.43	122.22	146.66	132.4	158.88	142.59	171.10
	.35	97.22	116.66	108.02	129.62	118.82	142.58	129.62	155.54	140.43	168.51	151.23	181.47
	.375	104.16	124.99	115.74	138.88	127.31	152.77	138.88	166.65	150.46	180.55	162.03	194.43
60 inches........	.1875	46.87	56.24	52.08	62.49	57.29	68.74	62.5	75	67.7	81.24	72.91	87.49
	.21	52.5	63	58.33	69.99	64.16	76.99	69.99	84	75.83	90.99	81.66	97.99
	.23	57.5	69	63.88	76.65	70.27	84.32	76.66	91.99	83.05	99.66	89.44	107.32
	.25	62.5	75	69.44	83.32	76.38	91.65	83.33	99.99	90.27	108.32	97.22	116.66
	.26	65	78	72.22	86.66	79.44	95.32	86.66	103.99	93.88	112.65	101.11	121.33
	.29	72.5	87	80.55	96.66	88.61	106.33	96.66	115.99	104.72	125.66	112.77	135.32
	.3125	78.12	93.74	86.8	104.16	95.48	114.57	104.18	124.99	112.95	135.54	121.52	145.82
	.33	82.5	99	91.66	109.99	100.83	120.99	109.99	132	119.16	142.99	128.33	153.99
	.35	87.5	105	97.22	116.66	106.94	128.32	116.66	139.99	126.38	151.65	136.11	163.33
	.375	93.75	112.5	104.16	124.99	114.58	137.49	125	150	135.41	162.49	145.83	174.99
66 inches........	.1875	42.61	51.13	47.34	56.8	52.07	62.49	56.81	68.17	61.55	73.86	66.28	79.53
	.21	47.72	57.26	53	63.63	58.33	69.99	63.63	76.35	68.93	82.71	74.24	89.08
	.23	52.27	62.72	58	69.69	63.88	76.65	69.69	83.62	75.5	90.6	81.31	97.57
	.25	56.81	68.17	63.13	75.75	69.44	83.32	75.75	90.90	82.07	98.48	88.38	106.06
	.26	59.09	70.9	65.65	78.78	72.22	86.66	78.78	94.53	85.35	102.42	91.91	110.29
	.29	65.90	79.08	73.23	87.87	80.55	96.66	87.87	105.44	95.2	114.24	102.52	123.02
	.3125	71	85.2	78.91	94.69	86.89	104.16	94.69	113.62	102.58	123.09	110.47	132.56
	.33	75	90	83.33	99.99	91.66	109.99	99.99	120	108.33	129.99	116.66	139.99
	.35	79.56	95.47	88.38	106.05	97.22	116.66	106	127.27	114.89	137.86	123.73	148.47
	.375	85.22	102.26	94.69	113.62	104.16	124.99	113.62	136.34	123.1	147.72	132.57	159.08
72 inches........	.1875	39.06	46.87	43.4	52.08	47.74	57.28	52.08	62.49	56.42	67.70	60.76	72.91
	.21	43.75	52.5	48.6	58.33	53.47	64.16	58.33	69.99	63.19	75.82	68.05	81.66
	.23	47.91	57.49	53.24	63.88	58.56	70.27	63.88	76.65	69.21	83.05	74.53	89.43
	.25	52.08	62.49	57.87	69.44	63.65	76.38	69.44	83.32	75.22	90.26	81.01	97.21
	.26	54.16	64.99	60.18	72.21	66.2	79.44	72.22	86.66	78.24	93.88	84.25	101.10
	.29	60.41	72.49	67.12	80.54	73.84	88.60	80.55	96.66	87.26	104.71	93.98	112.77

	.3125	65.10	78.12	72.33	86.8	79.57	95.48	86 8	104 16	94.03	112 83	101.27	121 52
	.33	68.75	82.5	76.38	91.65	84.02	100.82	91 66	109 99	99.3	119.16	106.94	128 32
	.35	72.91	87.49	81.01	97.21	89.11	106.93	97.22	116 66	105.32	126.38	113 42	136.1
	.375	78.12	93.74	86.8	104.16	95.48	114.57	104.16	124.99	112.84	135.43	121 52	145 82
78 inches........	.1875	36.05	43.21	40.06	48.07	44.07	52.87	48.07	57.68	52.08	62.49	56.08	67.29
	.21	40.38	48 45	44.87	53.84	49.35	59.22	53.84	64.60	58.33	69.99	62 82	75.38
	.23	44.23	53.07	49.14	58.96	54.05	64.86	58.95	70.76	63.88	76.65	68.80	82.56
	.25	48.07	57.68	53.41	64.09	58.76	70 5	64.4	76.92	69 44	83.32	74.78	89.73
	.26	50	60	55 55	66.66	66.11	73.33	66.66	79 99	72 22	86.66	77.77	93.32
	.29	55.76	66.91	61.96	74.35	68.16	81.79	74.35	89 22	80.55	96.66	86.75	104.1
	.3125	60.09	72.1	66.77	80.12	73 45	88.14	80.12	96.14	86.8	104.16	93.48	112.17
	.33	63.46	76.15	70.51	84.61	77.56	93.07	84.61	101.53	91.66	109.99	98 71	118.45
	.35	67.3	80.76	74 78	89.73	82 26	98.71	89.74	107.68	97.22	116.66	104.70	125.64
	.375	72.11	86.53	80.12	96.14	88.14	105.76	96.15	115.38	104.16	124.99	112.17	134.6
84 inches........	.1875	33 48	40.17	37.2	44 68	40.92	49.1	44.64	53.56	48 36	58.03	52.08	62.49
	.21	37.5	45	41.66	49.99	45.83	54.99	50	60	54.16	64.99	58.33	69.99
	.23	41.02	49.22	45.63	54 75	50.19	60.22	54.75	65.71	59.32	71.18	63.88	76 66
	.25	44.64	53.56	49.6	59.52	54.56	65.47	59.52	71.42	64.48	77.37	69 44	83.32
	.26	46.42	55.7	51.58	61.89	56.74	68 08	61.9	74.28	67.05	80 46	72 22	86.66
	.29	51.78	62.13	57.53	69.03	63.29	75.94	69.04	82.84	74.8	89.76	80.55	96.66
	.3125	55.8	66.96	62	74.4	68 2	81.84	74.4	89.28	80.6	96.72	86 8	104.16
	.33	58 92	70 7	65.47	78.56	72.02	86.42	78.57	94.28	85.11	102.13	91.66	109.99
	.35	62.5	75	69.44	83.32	76.38	91.65	83.33	99.99	90.27	108.32	97.22	116.66
	.375	66.96	80.35	74.4	89.28	81.84	98.2	89.28	107.13	96 72	116.06	104.16	124.99
90 inches........	.1875	31.25	37.5	34.72	41.66	38.19	45.82	41.66	49.99	45.13	54.15	48.68	58.33
	.21	35	42	38.88	46.65	42.77	51.32	46.66	55.99	50.55	60.66	54.44	65.32
	.23	38.33	45.99	42.59	51.10	46 85	56.22	51.11	61.33	55 37	66.44	59 62	71.54
	.25	41.66	49.99	46 29	55.54	50.92	61.1	55.55	66.66	60.18	72.21	64 81	77.77
	.26	43.33	51.99	48.14	57.76	52.96	63.55	57.77	69 32	62.59	75.1	67 4	80 88
	.29	48.33	57.99	53.7	64.44	59.07	70.8	64.44	77.32	69.81	83.77	75.18	90.21
	.3125	52.08	62.49	57.86	69.43	63.65	76.38	69.44	83 32	75.23	90 27	81.01	97.21
	.33	55	66	61.11	73.33	67.22	80 66	73.33	87.99	79.44	95.32	85.55	102.66
	.35	58.33	69.99	64.81	77.77	71 29	85 54	77.77	93.32	84.25	101.1	90.72	108.88
	.375	62 5	75	69.44	83.32	76.38	91.65	83.33	99 99	90.27	108 32	97.22	116 66
96 inches........	.1875	29.29	35.14	32.55	39 06	35.8	42.96	39.06	46 87	42.31	50.77	45.57	54.68
	.21	32.81	39.37	36.45	43.74	40.1	48.12	43.75	52.5	47 39	56 86	51.04	61.24
	.23	35.93	43.11	39.93	47.91	43 92	52.7	47.91	57.49	51.9	62 28	55.9	67.08
	.25	39.06	46.87	43.4	52.08	47 74	57.28	52.08	62.49	56.42	67.67	60.76	72.91
	.26	40.62	48.74	45.14	54.16	49.65	59.58	54.16	64.99	58.78	70.53	63 19	75.82
	.29	45.31	54.37	50.34	60.4	55.38	66.45	60.41	72.49	65.45	78.54	70.48	84.57
	3251	48.82	58.58	54.25	65.1	59.67	71.6	65.1	78 12	70.52	84.62	75.95	91.14
	33	51.56	61.87	57.29	68.74	63.02	75 62	68.75	82.5	74.47	89.36	80 2	96 24
	.35	54.68	65.61	60.76	72.91	66.83	80.19	72 91	87.49	78.99	94.78	85.06	102 07
	.375	58.58	70 29	65.1	78.12	71.61	85.93	78 12	93.74	84.63	101.55	91.14	109.6

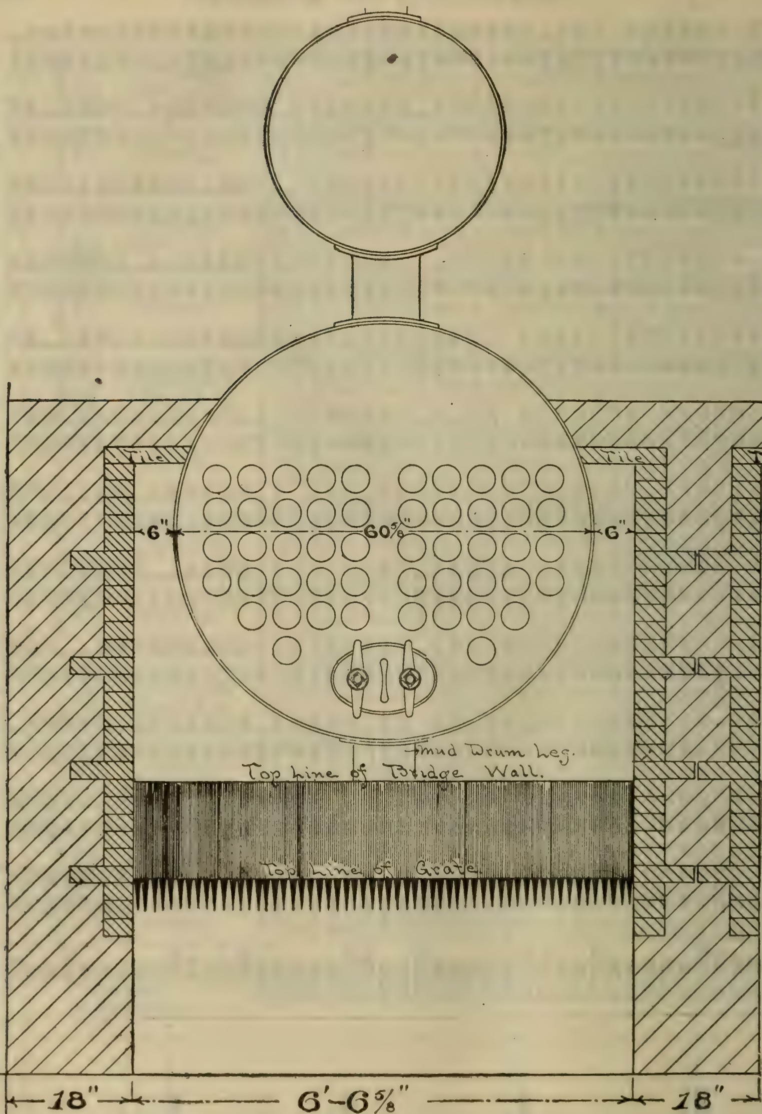


Fig. 271. Showing the proper place for closing in the boiler on the side — also the space between side of boiler and side walls.

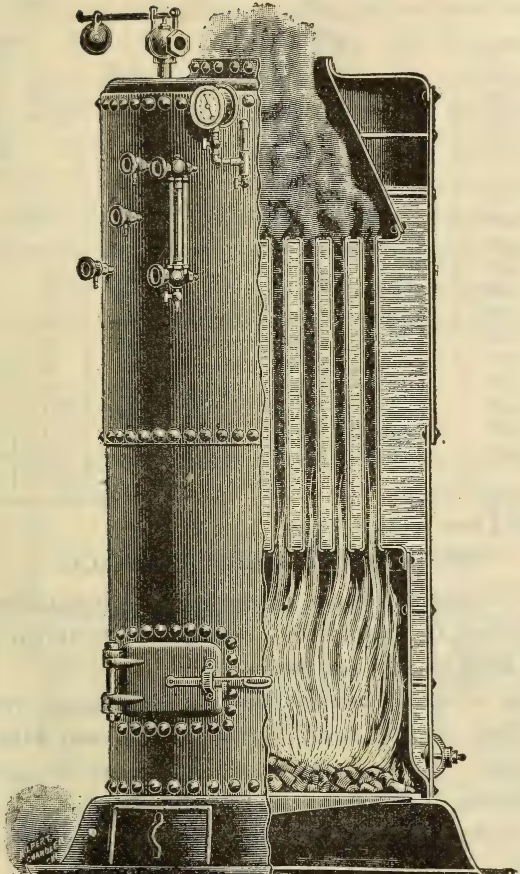


Fig. 272. Showing the proper place for gauge-cocks in a submerged tube boiler.

THE AMOUNT OF MATERIAL REQUIRED TO BRICK UP BOILERS OF DIFFERENT SIZE.

Size of Boiler.	Thickness of Walls.	Red Brick.	Fire Brick. Fire Clay ½ lb. to each brick.	Lime.	S. Tile. 6″ Tile.	Sand. Yds.	Cement for concrete footing 2 ft. and 2 ft. thick under all walls.
72″x22′	18″	10,500	2,500	18 bu.	88	8	9 bbl.
72″x20′	18″	10,000	2,300	18 bu.	80	8	8 bbl.
72″x18′	18″	9,500	2,200	17 bu.	72	7	8 bbl.
60″x20′	18″	9,500	2,200	17 bu.	80	7	8 bbl.
60″x18′	18″	9,000	2,000	16 bu.	72	7	8 bbl.
54″x20′	18″	8,700	1,900	15 bu.	80	6	8 bbl.
54″x18′	18″	8,000	1,800	15 bu.	72	6	8 bbl.
54″x16′	18″	7,500	1,700	14 bu.	64	6	7 bbl.
48″x18′	18″	7,500	1,600	14 bu.	72	6	7 bbl.
48″x16′	18″	7,200	1,500	14 bu.	64	5	7 bbl.
42″x18′	18″	7,000	1,400	12 bu.	72	5	7 bbl.
42″x16′	18″	6,500	1,300	12 bu.	64	4	7 bbl.

If 13″ wall ¼ less on Red Brick.

THE DOWN DRAUGHT FURNACE.

The down draught furnace is known as being one of the best smoke preventing furnaces in the market, while at the same time the cheapest kind of coal can be used.

The down draught furnace makes a good smoke record, even with overworked boilers, doing variable work, and with a marked economy in fuel. All experience with the down draught furnace, seems to indicate that smoke from boiler furnaces can now be abated by practical means, without hardship, no matter what the type of boiler.

Directions for firing the down draught furnace. — When firing the furnace, throw the coal evenly over the entire grate surface, from 6 to 8 inches in depth, a little heaviest at the rear end of the furnace. Do not put in too much coal — burn more air and economize fuel if possible, and

do not pile up the coal in front near the door. Never fire any fresh coal on the lower grates; let in air below the lower grates. When poking the fire, run the slice-bar down between the water grates and back the full length of the grates; then raise the slice-bar and gently shake the coal, and then pull it out without stirring up the fire. Never turn the fire over so that black coal gets down upon the water grates, unless there is a large clinker to remove. Never give the top grates a general cleaning, so as to leave a portion of the grates uncovered and the remainder with a hot fire on them, as this causes an uneven expansion in the different tubes forming the water grates, and is liable either to bend the tubes or strip off the threads where they enter the drums. When the top fire becomes clogged with clinkers so that it is hard

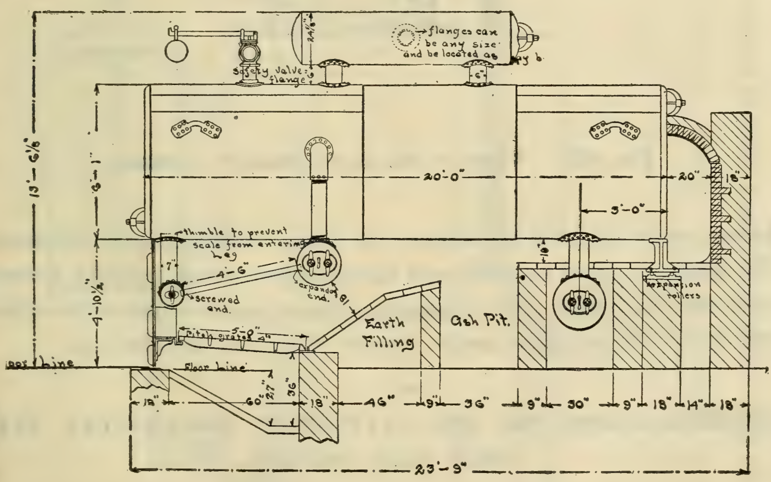


Fig. 273. Down draught furnace.

to keep up steam, run in the slice-bar and raise the clinkers to the top of the fire; remove the large clinkers, leave the small ones alone, and put on a fresh fire. The lower grates must have proper

attention. The coals must be raked over evenly and all holes filled up, particular care being taken that the grates are perfectly covered all over. If considerable coals have accumulated on the

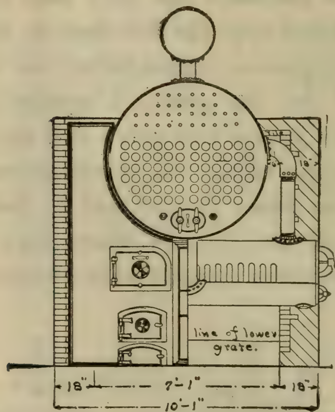


Fig. 274. View of the down draught furnace.

lower grates and the air spaces are closed with ashes or clinkers, the slice-bar must be used and the clinkers raised up and turned over and the larger ones removed. It is best to remove the clinkers every two or three hours, leaving the coals to burn up.

SPECIFICATIONS FOR ONE SIXTY-INCH HORIZONTAL SIX-INCH FLUE BOILER.

General directions. — There will be one boiler 20 feet long from out to out of heads and 60 inches inside diameter.

Material, quality, thickness, etc. — Material in shell of the above named boiler to be made of homogeneous flange steel $\frac{5}{16}''$ thick, having a tensile strength of not less than 60,000 lbs. to

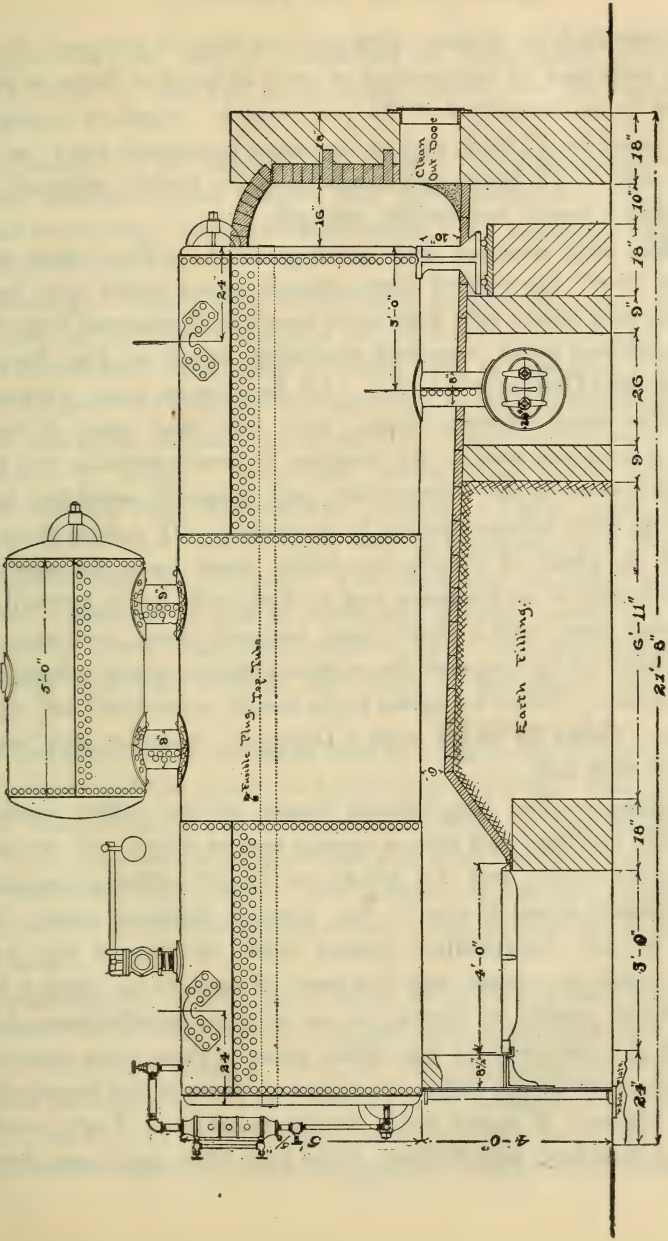


Fig. 275. Showing a good form of furnace for a long boiler.

the square inch of section, with not less than 56 per cent ductility, as indicated by contraction of area at point of fracture under test, or by an elongation of 25 per cent in length of 8 inches. Heads must be $\frac{1}{2}''$ thick and of the same quality of steel as that in the shell. All plates and heads must be plainly stamped with the maker's name, and tensile strength.

Tubes, size, number and arrangement. — The boiler must contain 18–6″ lap-welded flues, riveted to the heads with ten $\frac{1}{2}''$ rivets in each head ; said flues must be made of charcoal iron of the best American make, standard thickness, equal to the National Tube Works Company's make. All flues must have at least 3 inch clear space between them, and not less than 3 inches between flues and shell. All flanging of heads must be free from flaws or cracks of any description, and properly annealed in an annealing oven before riveting to the boiler. If 4-inch flues are wanted in place of 6-inch, the boiler must have 44 best lap-welded tubes, 4″ in diameter and 20 feet long, set in vertical and horizontal rows, with a clear space between them, vertically and horizontally of $1\frac{1}{4}''$, except the central vertical space, which is to be 4 inches. Holes for tubes to be neatly chamfered off on the outside. Tubes to be set with a Dudgeon expander, and beaded down at each end.

Riveting. — The longitudinal seams of the boiler must be above the fire line, and have a TRIPLE row of rivets ; all rivets to be $\frac{3}{4}''$ in diameter ; and all rivets to be of sufficient length to form upheads equal in size to the pressed heads of same. The rivets in the longitudinal seams must be spaced $3\frac{1}{4}''$ apart from center to center, and the rows of same to be pitched $2\frac{3}{16}''$ apart from center to center, so as to give an efficiency of the joint of $\frac{76}{100}$ per cent of the solid plate. Transverse seams to be single riveted with same size rivets as those in the longitudinal seams pitched 2″ apart from center to center. Care must be taken in punching and drilling holes that they may come fair in

construction; the use of a drift-pin to bring blind, or partially blind holes in line will be sufficient cause for the rejection of the boiler.

Calking. — The edges of the plates to be planed and beveled before making up the boilers, and the calking to be done with round nose tools, pneumatically driven; no split or wedge calking will be allowed.

Bracing. — There must be 22 braces in the boiler, one inch area at least, be nine above the flues on the front head and nine similar ones on the back head, none of which shall be less than 3′ 6″ long, made of good refined iron and securely riveted to the heads; the other end to be extended to the shell of boiler and riveted thereto with two $\frac{7}{8}$″ rivets. Care must be exercised in the setting of them, so they may bear uniform tension. There must be two braces below flues, one on each side of manhead, and riveted to the heads with two $\frac{7}{8}$″ rivets. The back end of brace to be extended backward to side of shell and riveted thereto by means of two $\frac{7}{8}$″ rivets; and two braces in back end above flues, one on each side and riveted the same as the other two below flues.

Manholes. — The boiler to have two manholes of the Hercules or Eclipse pattern, same to be of size 10″ x 15″, one located in front head, beneath the flues, and the other in rear head above the flues, and each to be provided with a lead gasket, grooved lid, two yokes and two bolts. The proportion of the whole to be such as will leave it as strong as any other portion of the head of like area.

Steam drum. — The boiler must be provided with one steam drum 30″ in diameter by 5′ in length, shell plates of which are to be $\frac{5}{16}$″ thick and heads $\frac{1}{2}$″ thick, of the same quality of material as that in the boiler. The heads must be bumped to a radius so as to give as near as practicable equal strength as to that in the shell without bracing. The longitudinal seams of the drum are

to be double riveted with $\frac{11}{16}''$ diameter rivets, pitched $2\frac{7}{8}''$ apart from center to center, so as to give an efficiency of the joint of 74 per cent of the solid plate.

Manhole in drum. — The drum must be provided with Hercules or Eclipse patented manhole, same to be of size 10″ x 15″, located in the center of one head, and to be provided with a grooved lid, lead gasket, two yokes and two bolts. The proportion of the whole to be such as will leave it as strong as any other portion of the head of like area.

To attach to boilers. — The steam drum must be attached to the boiler by means of two flange steel connecting legs, 8″ in diameter by 12″ in length, and securely riveted to boiler and steam drum shell.

Mud drum. — Boiler must be provided with one mud drum 24″ in diameter and of sufficient length so that each end may come flush with the outside of the boiler walls on each side; the quality and thickness of steel to be the same as that specified for the steam drum, and all seams to be single riveted; said mud drum to be provided with one Hercules or Eclipse patent manhole in one end, and to be of size 9″ x 14″, supplied with a grooved lid, lead gasket, two yokes and two bolts.

To attach to boiler. — The mud drum is to be attached to boiler by means of 8″ diameter steel connecting leg, about 16″ in length, properly riveted to boiler and mud drum shells.

Flanges. — The boiler to have one 8″ wrought steel flange riveted on top of steam drum; one wrought steel flange 4″ in diameter, about 5 feet from front end of boiler for safety valve one 2″ wrought steel flange on after end of boiler over the center of mud leg for supply pipe — all flanges to be threaded; 2″ hole in mud drum for blow-off; also 2 $1\frac{1}{4}''$ holes, one on top of boiler and one on end near bottom of boiler for water column.

Fusible plugs. — To have two fusible plugs; one inserted in shell from inside on second sheet, or about 5′ from forward end, 1

inch above flues; one plug inserted in top of flue, not more than three feet from after end.

Trimmings. — Furnish one 4″ spring or dead weight safety valve, 4″ diameter; one water combination column; provide same with two $1\frac{1}{4}$″ valves for the steam and water connections between the boiler and column, and one $\frac{1}{2}$″ valve for blow-pipe; said blow-pipe to be connected with ashpit; said combination barrel to be 4″ diameter, 18″ long, and made of cast-iron. Also, furnish one water gauge having a $\frac{3}{4}$″ x 15″ Scotch glass tube, bodies polished with wood wheels and guards, rods, bodies threaded $\frac{3}{4}$″; three gauge cocks $\frac{3}{4}$″ register pattern, polished brass bodies; one steam gauge with 10″ dial; one 2″ brass feed valve with 2″ check valve; one 2″ gate valve for blow-off from mud drum; also one asbestos packed stop-cock for same, so as to insure against the possibilities of a leak through the blow-pipe. Water column to have crosses in place of ells. Crosses to have brass plugs.

Castings, grates, doors, etc. — The boiler must be provided with a heavy three-quarter fire front of neat design, having double firing and ashpit doors, anchor bolts for anchoring fire fronts in place, heavy deadplates, a full set of fire liners 9″ deep for supporting firebrick on end, front and rear bearing bars; a full set of ordinary grate bars 4 ft. long, soot door and frame for cleaning out rear ashpit; a full set of skeleton arch plates; 12 heavy buck staves $9\frac{1}{2}$′ long, provided with tie rods, nuts and washers, heavy back stand with plate and expansion rollers; also furnish wrought plates to cover mud drum.

Fire tools. — Furnish in addition to above two sets of fire tools consisting of two pokers, two hoes, two slice-bars, two claws, and one six-inch flue brush with $\frac{1}{2}$″ pipe for handle.

Breeching. — Boiler must have a breeching fitted to front head and fastened thereto by means of bolts, stays and suitable pieces of angle iron, bent to conform to circle of boiler. The underside of breeching is to run across the head between the lower flues and

the manhole, leaving the manhole freely exposed; the sides of breeching are to be made of $\frac{3}{16}''$ steel, the front and doors of $\frac{1}{8}''$ steel; said doors to be hung by means of strap hinges, provided with suitable fastenings so as to give free access to all flues when open.

Uptake and damper. — An uptake having an area of 1221 square inches must be fitted to top of breeching. Said uptake must be of convenient form for attaching to a stack 40″ in diameter and provided with a close-fitting damper having a steel hand attachment, so that same may be operated conveniently from the boiler room floor.

Smoke stack. — There is to be provided for the above boiler one smoke stack 40″ in diameter by 90 feet in height, half of which is to be made of No. 8, and the other half of No. 10 best black sheet steel throughout, and supplied with two sets of four guy rods, each consisting of $\frac{3}{8}''$ galvanized wire cable guy strand with turn-buckles for same.

In general. — The above-mentioned boiler must be made of strictly first-class material and workmanship throughout, and subjected to a hydrostatic pressure of 150 pounds to the square inch before leaving the works of the manufacture.

Painting boiler breeching. — Smoke stack and boiler front, steam and mud drum, and all trimmings, to have two good coats of coal tar.

Masonry. — Boiler to be set in good substantial masonry, of hard burned brick and good mortar, made of clean, sharp sand and fresh burned lime. Walls to be 18″ thick. The outside walls to be laid up of selected hard burned brick, with close joints struck smooth and rubbed down. The sides, end and bridge walls, and boiler front, to have a foundation of 24″ wide and 12′ deep, laid in Portland cement. The ash pit to be paved with hard burned brick set on edge firmly, imbedded in Portland cement. For a distance of seven feet in front of the boiler and

continuing across entire width of front of boiler setting to be paved with hard burned brick set on edge, firmly imbedded in sand. The walls to be carried up to the full height and a row-lock course of brick 4″ thick to be carried over top of boiler from side wall to side wall, extending the whole length of boiler, and the entire arch to be plastered over on the outside with mortar. The bridge walls to be 24″, carried up to within 6″* of under side of boiler. The top of bridge wall to be of fire brick and made in the form of an inverted arch, conforming to the shell of the boiler. The space under boiler and back of bridge wall to the back end of boiler, to be filled in with earth or sand and the top paved with brick, and tapering from bridge wall back to back end to 12″* at back end, and in a similar form and shape, that is, inverted arch. The uptake for returning the smoke and heat at back end of boiler, to be arched over from rear wall against the back head of boiler 2″ above the tubes, the arch being made of arch fire brick, and backed up with red brick. Furnace to be lined throughout with first quality fire brick, dipped in fire clay with close joints and fire brick rubbed to place, from a point 2″ below grates, to where it safes in against boiler, and to be continued fire brick as far back as the rear end of setting and across rear end of same; it being the intent that all interior surfaces of the setting with which the heat comes in contact, shall be faced with fire brick. Every sixth course to be a header course.

Smoke connections. — The connection from boiler to chimney to be made of No. 12 black iron, with cleaning door and damper in same.

BANKING FIRES.

Different engineers pursue different methods in banking fires. One method is to push the fire back one-third towards the bridge wall, and clean off the grate in front. Then shovel in from 150 to 300 lbs. of fine coal on top of the fire, closing ash-pit doors

* These distances should be doubled for bituminous coal.

and leaving furnace doors open, with damper open enough to let the gases escape. Others bank after this fashion but close all doors and air holes, leaving the damper partially open. Another method is to level the fire all over the grate, and shovel in from 150 to 500 lbs. of fine coal, — depending on the size of the grate, — and then cover the whole surface with wet ashes to a good depth, so that no fire nor flame can be seen, then close the ash-pit doors, leaving the furnace doors ajar, and leave the damper partially open so that the gases may escape. In the morning, rake out the ashes, clean the fire, and throw in fresh coal.

INSTRUCTIONS FOR BOILER ATTENDANTS.

The following instructions apply more particularly to horizontal return tubular boilers, although in a general way they are applicable to all types of boilers.

Never start a fire under a boiler until you are positively certain that there is sufficient water in the boiler, — at least two gauges of water. Do not trust to the water gauge alone, but try the gauge cocks also, and try them at intervals during the day, because the water gauge pipe connections may be choked and cause a false water level.

Before starting a fire be sure that the blow-off cock is closed and not leaking.

Before it is time to start the engine, pump up three gauges of water, and blow off one gauge, in order to get rid of mud and other sediment. If the boiler has a surface blow-off, — commonly called a "skimmer," — blow off the scum before stopping the engine for the day.

When the day's work is done, leave three gauges of water in the boiler, to allow for leakage and evaporation during the night.

Never raise steam hurriedly. Sudden changes of temperature may produce fractures, or start leaks.

In starting a fire in a furnace, a good plan is to cover the grate with a thin layer of coal and to place the shavings and wood on the coal and then light the shavings.

The advantage of placing a covering of coal on the grate before the wood and shavings, is that it is a saving of fuel, as the heat that would be transmitted to the bars is absorbed by the coal, and the bars are also protected from the extreme heat of the fresh fire.

Lift the safety-valve, — if of the lever pattern, — every morning while raising steam, and satisfy yourself that it is in good working order, and that the ball is set at the proper point on the lever. The most disastrous explosions have occurred with boilers whose safety-valves had been stuck down or overloaded.

Keep the boiler shell free of soot. Soot is a very good non-conductor of heat, and considered worse than scale inside of a boiler.

Keep your boiler tubes free from soot and dust. Choked tubes impair the draft. The tubes should be cleaned twice a week, or oftener.

Soot collects also in a stack or chimney and in the connection between the breeching and stack, and interferes with the draft.

Open your boiler every two weeks, or, as often as necessary, — depending on the kind of feed-water used, — and clean out the mud and scale. At the same time examine all of the stays, and see that they are taut and in good order. Also, look for pitting around the mud-drum connection, and for grooving in the side seams. Examine all outlets and pipe connections, and look for indications of "bagging" in the furnace sheets.

Clean off the fusible plugs both inside and outside of the boiler. A fusible plug covered with soot on the fire side, and with scale on the water side, is no longer a "safety plug." Renew the filling in safety plugs, at least once a year. They are filled with pure Banca tin.

Be perfectly satisfied that your boiler is in good condition internally before you close it up.

Just as soon as you have fastened the man-head in its place, turn on the feed-water until you get at least three gauges of water. Fires have been built under empty boilers, and will be again, if you forget to turn on the feed water after cleaning out.

Do not empty a boiler while it is under steam pressure, but allow it to get cold before letting the water run out.

If you are in a great hurry and can't wait for the boiler to cool down, nor for the brickwork or anything else to cool down, draw the fire and open the furnace and ash-pit doors, then turn on the feed water, and from time to time blow out, until the steam gauge shows no pressure; then shut off the feed-water, raise the safety valve, open the blow-off cock, then open up the boiler.

Before opening a man-hole, lift the safety-valve, so as to be sure that there is neither pressure nor vacuum in the boiler.

Look well after the brick-work surrounding your boiler, and stop all cracks in the walls with mortar or cement, as soon as discovered. They impede the draft, and cool the plates of the boiler, causing a waste of fuel.

See that the bridge wall is in perfect condition, because a gap in the bridge wall might cause a "bag" in the boiler by concentrating the flames on one spot.

Never allow any bare places on the grate, nor any accumulation of ashes, or dead coal in the corners of the furnace, as such places admit great quantities of cold air into the furnace, and render the combustion very imperfect.

In firing with anthracite coal, do not poke and stir up the fire, as with soft coal, but let it alone.

In firing soft slack coal, fire very lightly but frequently, carrying a thin fire.

In firing with soft lump coal, carry a thick fire, say from six to eight inches deep, according to the size of the furnace.

In firing up, you may spread the fresh coal evenly all over the grate, or, you may push the live coals back towards the bridge-wall, leaving a thin bed of live coals near the furnace doors, and spreading the fresh coal on top of it. This is called carrying a coking fire. Some prefer the one and some the other method of firing.

In case you should find the water in the boiler out of sight, and a heavy fire in the furnace, don't get rattled, and don't lose your head. Open the furnace doors, and close the ash-pit doors, and cover the fire with wet ashes, or damp clay, completely smothering it. Let everything else alone, including the safety valve and the engine. Now wait until the boiler cools down and the gauge shows no pressure, then turn on the feed-water.

On the other hand, if there is but very little fire in the furnace, you may draw the fire, instead of covering it with ashes or clay.

If your boiler foams badly and you are uncertain as to the water level, stop the engine, and the true water level will show itself at once.

If your boiler primes and water is carried over to the engine, it shows that there is want of sufficient steam room in the boiler. Either put a dry-pipe in the boiler, or, increase the steam pressure if the boiler will safely stand it.

Never attempt to calk a leaky seam in a boiler under steam pressure, because the jar caused by the hammer blows might cause a rupture of the seam. Better to be on the safe side always when repairs are required in a steam boiler, and wait until the boiler is cold. The above applies to steam pipes and valve casings, also.

Never open any steam valves suddenly, nor close them suddenly either, because it is highly dangerous to do so, particularly if there is considerable water in the pipes. The effect is the same as water hammer in water pipes.

Smoke is caused by too little air supply, or by the flames being

prematurely cooled. Therefore, after firing up with fresh coal, it might be necessary to leave the furnace doors ajar in order to supply sufficient air above the fuel.

Remember that it takes nearly 24 cubic feet of air for the proper combustion of one pound of soft coal. Hard coal does not require so much.

Each and every boiler in a battery should have its own independent safety-valve and steam gauge.

If you are obliged to force your fire, watch your furnace sheets for indications of "bagging," if the water space below the lowest row of tubes is cramped. Water-tube boilers are less liable to suffer from the effects of forced fires than shell boilers.

With an intensely hot fire under a shell boiler, the furnace sheets are liable to bag, unless there is ample water space between the shell of the boiler and the bottom row of tubes.

The use of mineral oil to remove or prevent boiler scale, is not to be recommended.

Have your feed water analyzed, and use a scale preventer adapted to its requirements.

By all means endeavor to secure a steady furnace temperature, and a steady steam pressure, for herein lies much economy of fuel. Fluctuations are wasteful.

Put a damper in your chimney and adjust it to the needs of your furnace. Try to prevail on your employer to put in a shaking grate. It will enable you to carry a steady furnace temperature, and also enable you to keep the air spaces in your grate free and open without breaking up your fire.

RULES AND PROBLEMS RELATING TO STEAM BOILERS.

To find the safe working pressure: —

U. S. Rule. — Multiply one-sixth ($\frac{1}{6}$) of the lowest tensile strength found, stamped on any plate in the cylindrical shell,

by the thickness — expressed in inches or parts of an inch — of the thinnest plate in the same cylindrical shell, and divide by the radius or half diameter — also expressed in inches — and the result will be the pressure allowable per square inch of surface for single riveting; to which add 20 per cent for double riveting, when all the holes have been "fairly drilled" and no part of such hole has been punched.

A. S. of M. E. Rule. — First, find the tensile strength of the solid plate between the centers of two adjacent rivet holes. Call this factor A.

Next, find the tensile strength of the solid plate between the centers of two adjacent rivet holes, less the diameter of one rivet hole. Call this factor B.

Next, find the shearing strength of the rivets. Call this factor C.

Now divide whichever is the smaller factor B or C by A, and the quotient will give the strength of the joint as compared with the solid plate — expressed as a percentage. Then multiply the tensile strength of the plates by the thickness of plates — in fractional parts of an inch — and multiply this product by the percentage as found above, and divide this last product by the radius of the shell in inches, and the quotient will be the bursting pressure.

Divide this quotient by the factor of safety and the result will give the safe working pressure.

Example. — What is the safe working pressure for a steel boiler 60 inches in diameter, with side seams double riveted, tensile strength of plates 60,000 lbs. per sqr. in., thickness of plate $\frac{3}{8}$ inch. Diameter of rivet holes $\frac{15}{16}$ inch, pitch of rivets $3\frac{1}{4}$ inches, shearing strength of rivets 38,000 lbs. per sqr. in., and factor of safety 5?

Ans. By U. S. rule, 150 lbs. per sqr. in.
By A. S. of M. E. rule, $106\frac{1}{2}$ lbs. per. sqr. in.

Operation by U. S. rule: —

$$\frac{60,000}{6} = 10,000. \quad \text{And, } 10,000 \times \tfrac{3}{8} = 3750.$$

And, $\frac{3750}{30} = 125$. And, $125 \times .20 = 25$.

Then, $125 + 25 = 150$.

Operation by A. S. of M. E. rule: —

$\frac{3}{8}'' = .375''$.

$\frac{15}{16}'' = .9375''$.

Then, $60,000 \times 3\frac{1}{4} \times .375 = 73,125$ lbs., the strength of the solid plate between the centers of two adjacent rivet holes. Call this factor A. Also, $3\frac{1}{4} = 3.25$.

Then, $3.25 - .9375 = 2.3125$.

And, $60,000 \times 2.3125 \times .375 = 52,031.25$ lbs. the strength of the plate between two adjacent rivet holes. Call this factor B.

Then, $.9375 \times .9375 \times .7854 = .69029$ of a square inch, the area of one rivet hole. There are two rows of rivets.

Then, $.69029 \times 2 = 1.38058$ sqr. ins. the area of two rivet holes combined.

Then, $38,000 \times 1.38058 = 52,462.04$ lbs., the resistance of rivets to shearing. Call this C. Now since B is less than C, divide 52,031.25 by 73,125 and get as a quotient .71 +, thus showing the strength of the joint to be more than 71 per cent of the strength of the solid plates.

Then, $\frac{60,000 \times .375 \times .71}{30} = 532.5$ lbs. per sqr. in., the bursting pressure.

And, $\frac{532.5}{5} = 106.5$ lbs. per sqr. in., the safe working pressure.

To find the horse power of a horizontal return tubular boiler, from its heating surface: —

Rule. — Find the heating surface in square feet, of the shell of the boiler, measuring from one fire line to the other. Next find the internal heating surface of all the tubes in square feet. Add the two results together and divide their sum by 12, and the quotient will be the H. P. approximately. The heads are omitted.

Example. — What is the H. P. of a horizontal return tubular boiler 60 inches in diameter and 20 feet long, with 44 four-inch tubes each 20 feet long, the distance from fire line to fire line being 9 feet? Ans. 86.65 H. P.

Operation. — The internal diameter of a 4-inch tube is 3.732 inches.

Then, $20 \times 9 = 180$ square feet of heating surface in the shell.

And, $\frac{3.732 \times 3.1416}{12} = .9770376$ ft., the circumference of one tube in feet.

And, $.9770376 \times 20 \times 44 = 859.793 +$ sqr. ft., the total heating surface of the tubes.

Then, $\frac{180 + 859.793}{12} = 86.65$ nearly.

To find the factor of evaporation: —

Rule. — From the total number of heat units in one pound of steam at the given pressure, subtract the number of heat units in one pound of the feed water at its given temperature, and divide the remainder by 965.7, which is a constant.

Example. — A boiler evaporates 6,000 lbs. of water per hour from feed water at 210 degrees into steam at 125 lbs. gauge pres-

sure, what is the equivalent evaporation "from and at," 212°? What is the H. P. of the boiler?

Ans. Equiv. evap. 6276 lbs.
H. P. 182, nearly.

Operation. — The total number of heat units in steam at 125 lbs. per sqr. in. gauge pressue is 1221.5351.

The number of heat units in feed-water at 210 degrees equals 210.874. The latent heat of steam at atmospheric pressure, equals 965.7.

Then, $1221.5351 - 210.874 = 1010.6611$.

And, $\frac{1010.6611}{965.7} = 1.046$, the factor of evaporation.

And, $6000 \times 1.046 = 6276$ the equivalent evaporation.

Then, $\frac{6276}{34.5} = 181.9$ H. P.

To find how many pounds of steam at a given absolute pressure will flow through an orifice of one square inch area in *one second:* —

Rule. — Divide the absolute pressure by the constant number 70.

Example. — How many pounds of steam at 85 lbs. per sqr. in. gauge pressure, will flow through an orifice one inch in diameter, in *one second?* Ans. 1.122 lbs.

Operation. — A hole 1 inch in diameter has an area of .7854 of a sqr. inch.

And $85 + 15 = 100$ lbs. absolute.

Then, $\frac{100 \times .7854}{70} = 1.122$.

The weight of a cubic foot of steam at 100 lbs. per sqr. in. absolute pressure is .2307 of a pound. Then, $\frac{1.122}{.2307} = 4.86 +$ cubic feet.

To find the width of a reinforcing ring for a round hole in a flat surface, when the ring must contain as many square inches as were cut out of the plate, and when the ring and the plate are of the same thickness: —

Rule. — Find the area of the hole in square inches and multiply it by 2. Divide this product by .7854 and extract the square root of the quotient for the diameter of the ring over all. Subtract the diameter of the hole from the diameter over all, and divide the remainder by 2 for the width of the ring.

Example. — What should be the width of a reinforcing ring for a hole 10 inches in diameter, the metal cut out, and the metal in the ring being $\frac{3}{8}$ in. thick? Ans. $2\frac{1}{16}$ inches.

Operation. — $10 \times 10 \times .7854 = 78.54$ sqr. ins. area of hole.

And, $78.54 \times 2 = 157.08$ sqr. ins. in both hole and ring.

And, $\dfrac{157.08}{.7854} = 200.$

And, $\sqrt{200} = 14.142 +.$

And, $14.142 - 10 = 4.142.$

Then, $\dfrac{4.142}{2} = 2.071''$ or practically $2\frac{1}{16}''$.

To find the width of a reinforcing ring for an elliptical manhole in a flat surface, when the ring must contain as many square inches as are contained in the hole, and the metal cut out and metal in the ring are of the same thickness: —

Rule. — Square the short diameter of the hole and add to it six times the short diameter multipled by the long diameter, and to this product add the square of the long diameter, and extract the square root of the sum. From this root subtract the sum of the short diameter added to the long diameter, and divide the remainder by 4 for the width of the ring.

Example. — What should be the width of a reinforcing ring for a manhole $11'' \times 15''$? Ans. $2\frac{1}{16}$ inches.

Operation. — $11'' \times 11'' = 121$.

And, $11 \times 15 \times 6 = 990$.

And, $15 \times 15 = 225$.

Then, $121 + 990 + 225 = 1336$.

And, $\sqrt{1336} = 36.551$.

And, $11 + 15 = 26$.

Then, $36.551 - 26 = 10.551$.

And, $\frac{10.551}{4} = 2.637 +$ ins. the width of the ring, or, practically $2\frac{11}{16}$ ins.

Then, $2.637 \times 2 = 5.274''$.

And, $11 + 5.274 = 16.274''$ short diameter of ring over all.

And, $15 + 5.274 = 20.274''$ long diameter over all.

Proof: $20.274 \times 16.274 \times .7854 = 259.13 +$ square inches area of hole and ring.

And, $15 \times 11 \times .7854 = 129.59 +$ sqr. ins. area of hole alone.

Then, $259.13 - 129.59 = 129.54$.

THE AMOUNT OF STEAM USED WITH VALVE OPEN WIDE, WITH STEAM JETS AS A SMOKE PREVENTIVE.

STEAM JETS.

Given two boilers with separate furnaces, having 4 steam jets in each furnace, and each jet $\frac{1}{16}$ inch in diameter, the steam pressure being 100 lbs. per sqr. inch by the gauge. How many pounds of steam at this pressure will flow through the 8 nozzles in 12 hours? Answer. 1739 lbs. nearly.

Operation: $\frac{1}{16}'' = .0625''$.

Then, $.0625 \times .0625 \times .7854 = .003067968750$ sqr. inch, area of 1 jet.

And, $.003067968750 \times 8 = .02454375$ sqr. inch, the combined area of 8 jets.

Also, $100 + 15 = 115$ lbs. per sqr. inch, the absolute steam pressure.

And, $\frac{115}{70} = 1.64$ lbs. of steam *per second* that will flow through an orifice of *1 square inch area.*

Then, $1.64 \times .02454375 = .04025175$ lbs. of steam *per second* flowing through the 8 jets.

Again: There are 43,200 seconds in 12 hours.

Thus: $12 \times 60 \times 60 = 43,200$.

Then, $.04025175 \times 43,200 = 1738.8756$ lbs. of steam will flow through 8 jets in 12 hours' time.

Taking a high speed automatic cut-off engine using 20 lbs. of steam per H. P. per hour, the 8 steam jets would waste enough steam in 12 hours to run —

A 10 H. P. engine for $8\frac{1}{2}$ hours.
A 20 " " " $4\frac{1}{4}$ "
A 40 " " " $2\frac{1}{8}$ "
An 80 " " " $1\frac{1}{16}$ "

Thus $10 \times 20 = 200$.

And, $\frac{1739}{200} = 8\frac{1}{2}$ nearly.

$20 \times 20 = 400$.

And, $\frac{1739}{400} = 4\frac{1}{4}$ nearly.

$80 \times 20 = 1600$.

And, $\frac{1739}{1600} = 1\frac{1}{16}$ nearly.

CHAPTER XIX

THE STEAM PUMP.

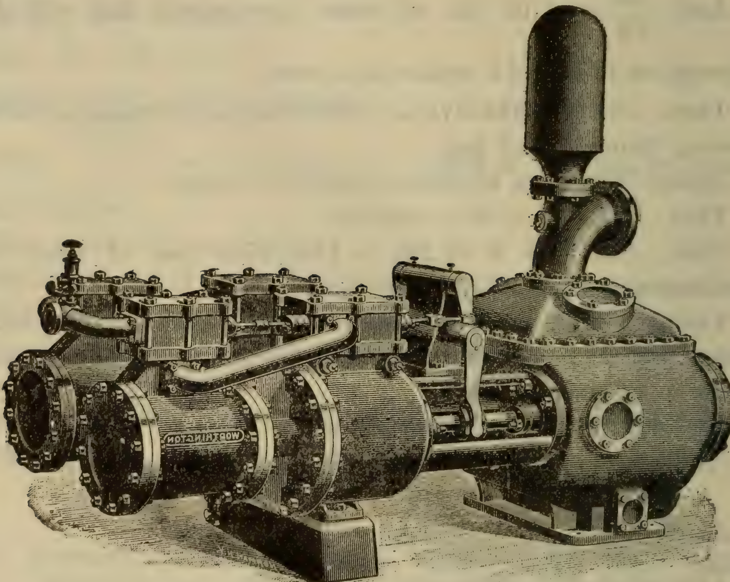


Fig. 276. The Worthington compound pump.

THE WORTHINGTON COMPOUND PUMP.

In the arrangement of steam cylinders here employed, the steam is used expansively, which cannot be done in the ordinary form. Having exerted its force through one stroke upon the smaller steam piston, it expands upon the larger during the return stroke, and operates to drive the piston in the other direction. This is, in effect, the same thing as using a cut-off on a crank engine, only with the great advantage of uniform and steady action upon the water.

Compound cylinders are recommended in any service where the saving of fuel is an important consideration. In such cases, their greater first cost is fully justified, as they require 30 to 33 per cent less coal than any high-pressure form on the same work.

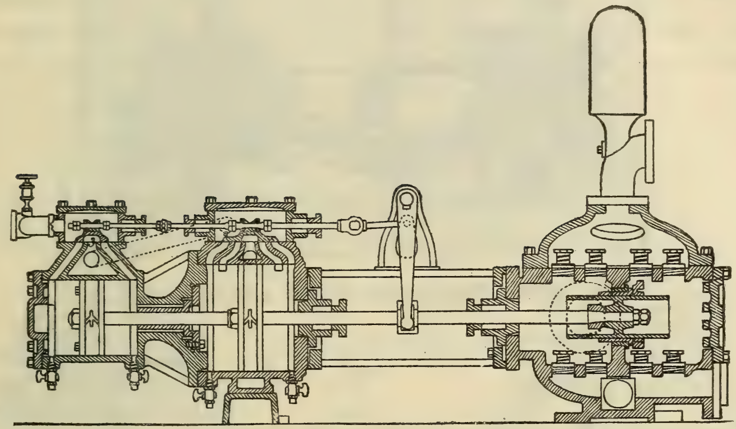


Fig. 277. Showing a sectional view of the Worthington compound pump.

This cut shows the steam valves properly set.

On the larger sizes, a condensing apparatus is often added, thus securing the highest economical results.

Any of the ordinary forms of steam pumps can be fitted with compound cylinders.

It should be remembered that, as the compounds use less steam their boilers may be reduced materially in size and cost, compared with those required by the high-pressure form. This principle of expansion without condensation cannot be used with advantage where the steam pressure is below 75 lbs.

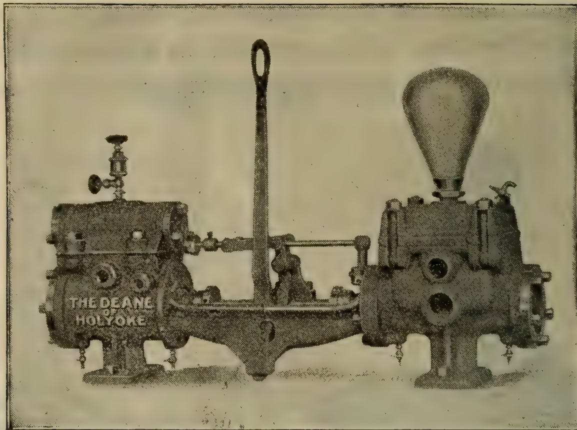


Fig. 278. The Deane direct acting pump.

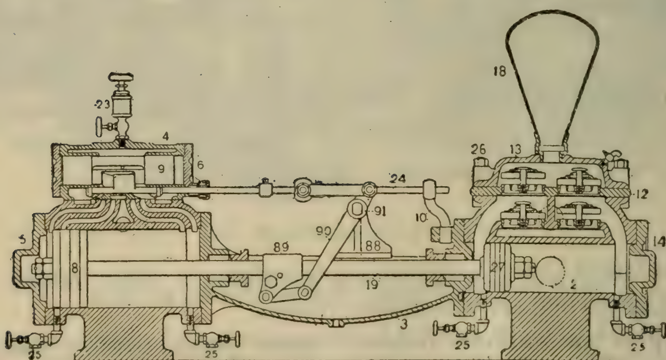


Fig. 279. Sectional view of the Deane pump.

DEANE DIRECT ACTING STEAM PUMP.

The operation of the steam valves.—In the Deane steam pump a rotary motion is not developed by means of which an

eccentric can be made to operate the valve. It is, therefore, necessary to reverse the piston by an impulse derived from itself at the end of each stroke. This cannot be effected in an ordinary single-valve engine, as the valve would be moved only to the center of its motion, and then the whole machine would stop. To overcome this difficulty, a small steam piston is provided to move the main valve of the engine. In the Deane steam pump, the lever *90*, which is carried by the piston rod, comes in contact

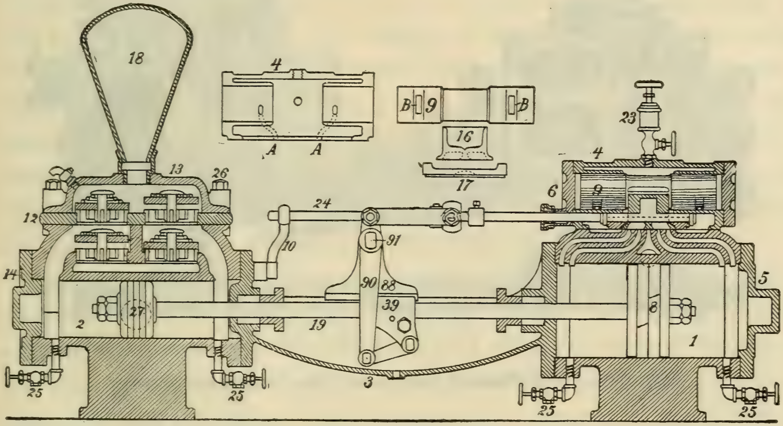


Fig. 280. Showing the valves properly set.

with the tappet when near the end of its motion, and by means of the valve-rod *24*, moves the small slide-valve which operates the supplemental piston *9*. The supplemental piston, carrying with it the main valve, is thus driven over by steam and the engine reversed. If, however, the supplemental piston fails accidentally to be moved, or to be moved with sufficient promptness by steam, the lug on the valve-rod engages with it and compels its motion by power derived from the main engine.

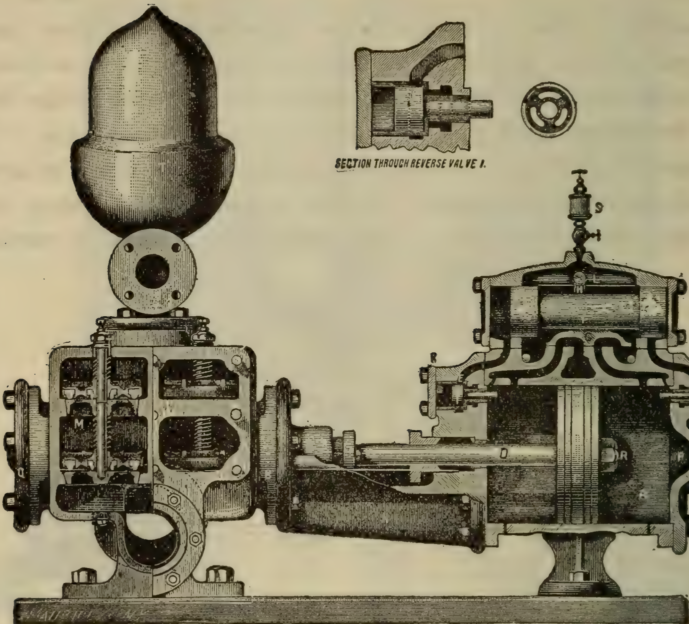


Fig. 281. Sectional view of the Cameron pump.

The above is a sectional view of the steam end of a Cameron pump.

Explanation: *A* is the steam cylinder; *C*, the piston; *D*, the piston rod; *L*, the steam chest; *F*, the chest piston or plunger, the right-hand end of which is shown in section; *G*, the slide valve; *H*, a starting bar connected with a handle on the outside; *I I* are reversing valves; *K K* are the bonnets over reversing valve chambers; and *E E* are exhaust ports leading from the ends of steam chest direct to the main exhaust, and closed by the reversing valve *I I*; *N* is the body piece connecting the steam and water cylinder.

Operation of the Cameron pump: Steam is admitted to the steam chest, and through small holes in the ends of the plunger; *F* fills the spaces at the ends and the ports *E E* as far as the reversing valves *I I;* with the plunger *F* and slide valve *G* in position to the right (as shown in cut), steam would be admitted to the right-hand end of the steam cylinder *A*, and the piston *C* would be moved to the left. When it reaches the reversing valve *I* it opens it and exhausts the space at the left-hand end of the plunger *F*, through the passage *E;* the expansion of steam at the right-hand end changes the position of the plunger *F*, and with it the slide valve *G*, and the motion of the piston *C* is instantly reversed. The operation repeated makes the motion continuous. In its movements, the plunger *F* acts as a slide valve to shut off the ports *E E*, and is cushioned on the confined steam between the ports and steam chest cover. The reversing valves *I I* are closed immediately the piston *C* leaves them, by pressure of steam on their outer ends, conveyed direct from the steam chest.

Operation. — Supposing the steam piston *C* moving from right to left: When it reaches the reversing valve *I* it opens it and exhausts the space on the left-hand end of the plunger *F*, through the passage *E*, which leads to the exhaust pipe; the greater pressure inside of the steam chest changes the position of the plunger *F* and slide valve *G*, and the motion of the piston *C* is instantly reversed. The same operation repeated at each stroke makes the motion continuous. The reversing valves *I I* are closed by a pressure of steam on their large ends, conveyed by an unseen passage direct from the steam chest. When a pump is first connected, remove the bonnets *K K* and valves *I I* and blow steam through to remove any dirt, oil or gum that may be lodged in the steam ports. Take valve *F*, valve *G* and *I I* out and wipe off with clean waste, and then oil and put back. Then see that the packing is not too tight. When a Cameron pump has been run a long time, the plunger *F* becomes worn and leaks enough steam to

cause the valve *F* to become balanced. The effect of this is, the pump will remain on the end; to overcome this, take out plunger *F*, or piston, as it is called by some, and drill the little hole that you will find in the ends of same a little larger, say about one-fourth larger; that will increase the pressure on both ends of plunger *F;* as soon as the piston comes in contact with valve *1* the steam is exhausted to exhaust pipe.

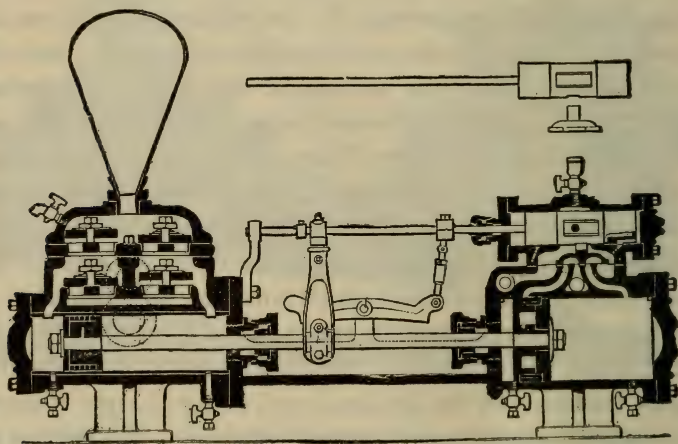


Fig. 282. Sectional view of the Knowles pump.

THE KNOWLES DIRECT ACTING STEAM PUMP.

Explanation of steam valves, etc. — The Knowles, in fact, all first-class direct acting steam pumps, are absolutely free from what is termed a "dead center," when in first-class order.

This feature in the Knowles pump is secured by a very simple and ingenious mechanical arrangement, *i. e.*, by the use of an auxiliary piston, which works in the steam chest and drives the main valve. This auxiliary or "chest piston," as it is called, is driven backward and forward by the pressure of steam, carrying

with it the main valve, which valve, in turn, gives steam to the main steam piston that operates the pump. This main valve is a plain slide valve of the *B* form, working on a flat seat. The chest piston is slightly rotated by the valve motion ; this rotative movement places the small steam ports, *D*, *E*, *F* (which are located in

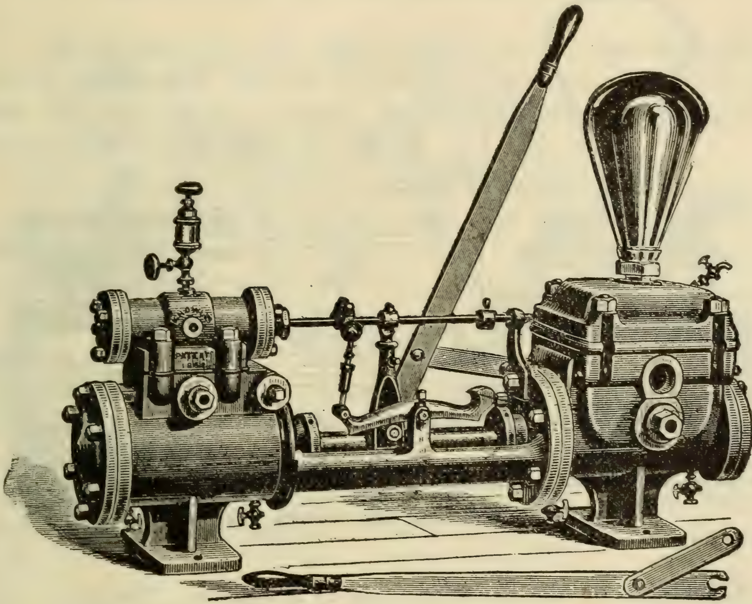


Fig. 283. The Knowles direct acting steam pump.

the under side of the said chest piston), in proper contact with corresponding ports *A B* cut in the steam chest No. 31. The steam entering through the port at one end and filling the space between the chest piston and the head, drives the said piston to the end of its stroke and, as before mentioned, carries the main slide valve with it. When the chest piston has traveled a certain distance, a port on the opposite end is uncovered and steam there enters, stopping its further travel by giving it the necessary

cushion. In other words, when the rotative motion is given to the auxiliary or valve-driving piston by the mechanism outside, it opens the port to steam admission on one end, and at the same time opens the port on the other end to the exhaust.

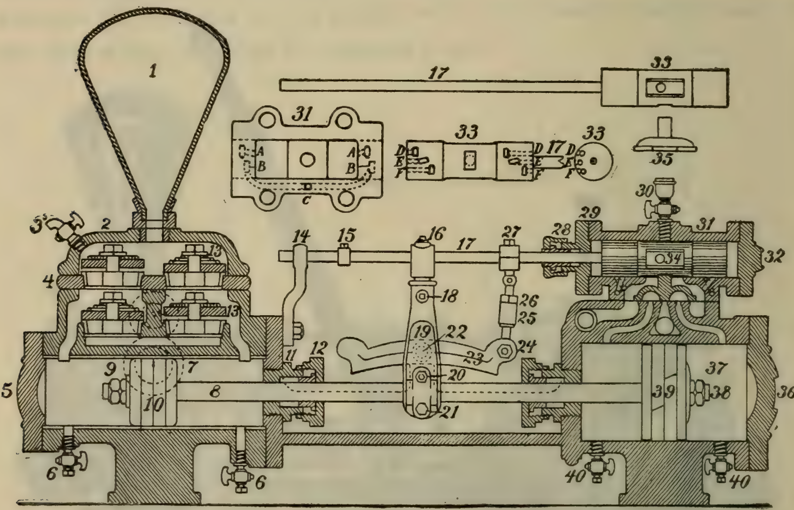


Fig. 284. Showing the valves properly set.

Operation of the Knowles pump is as follows: The piston rod, with the tappet arm, moves backward and forward from the impulse given by the steam piston. At the lower part of this tappet arm is attached a stud or bolt, on which there is a friction roller. This roller coming in contact with the "rocker bar" at the end of each stroke, operates the latter. The motion given the "rocker bar" is transmitted to the valve rod by means of the connection between, causing the valve rod to partially rotate. This action, as mentioned above, operates the chest piston, which carries with it the main slide valve, the said valve giving steam to the main piston. The operation of the pump is complete and

continuous. The upper end of the tappet arm does not come in contact with the tappets on the valve rod, unless the steam pressure from any cause, should fail to move the chest piston, in which case the tappet arm moves it mechanically.

ADJUSTMENT OF THE KNOWLES PUMP.

1. Should the pump run longer stroke one way than the other, simply lengthen or shorten the rocker connection (part *25*) so that rocker bar (part *23*) will touch rocker roller (*20*) equally distant from center (*22*).

2. Should a pump hesitate in making its return stroke, it is because rocker roller (*20*) is too low and does not come in contact with the rocker bar (*23*) soon enough. To raise it, take out rocker roller stud (*20A*), give the set screw in this stud a sufficient downward turn, and the stud with its roller may at once be raised to proper height.

3. Should valve rod (*17*) ever have a tendency to tremble, slightly tighten up the valve rod stuffing box nut (*28*). When the valve motion is properly adjusted, tappet tip (*16*) should not quite touch collar (*15*) and clamp (*27*). Rocker roller (*20*), coming in contact with rocker bar (*23*) will reverse the stroke.

Operation and construction of the

HOOKER DIRECT ACTING STEAM-PUMP.

The parts being in position, as shown, the steam on being admitted to the center of the valve chamber, brings its pressure to bear on the main and supplemental flat slide valve *4* and *7*, and also within the recess in the center of the supplemental piston *6*. The recess incloses the main valve *4*, so that this valve will move with the supplemental piston whenever the steam is supplied to

and exhausted from each end of this piston. The live steam passes through the left-hand ports A^1 B^1, driving the main piston *2* to the right, and the exhaust passes out through the right-hand ports *A* and *C* under the cavity in the main valve *4* to the atmosphere. As the main piston nears the right hand port, the valve lever *13*, which is attached to the piston rod *3*, brings the dog *17*, in plate *16*, in contact with the valve arm *15*, and moves the supplemental valve *7* to the right, thus supplying live steam to the

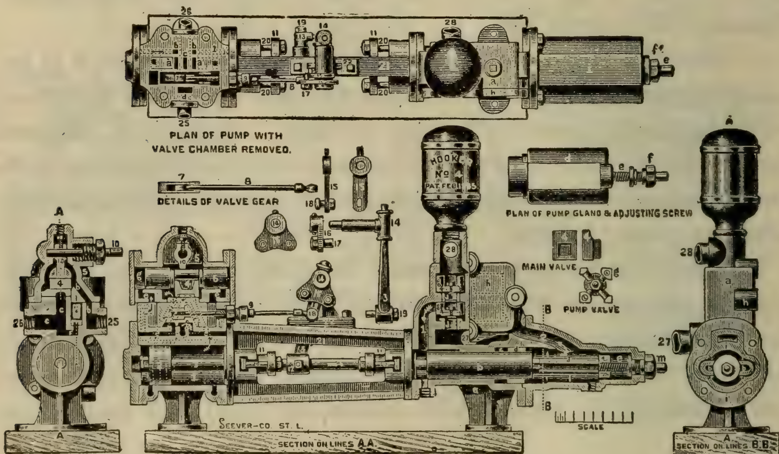


Fig. 285. Plan and sectional views of Hooker pump.

right of the supplemental piston *6*, and exhausting from the left through the ports *e e*. As the supplemental piston incloses the main valve, this valve is carried with it to the left. Steam now enters the right-hand ports *A B* and is exhausted from the left-hand main port *A*. The engine commences its return stroke and the operation just described becomes continuous. As the main piston (*2*) closes the main port (*A*) to the right, it is arrested on compressed exhaust steam. The main valve 4 having closed the auxiliary ports (*B*) leading to that end of the main cylinder, the

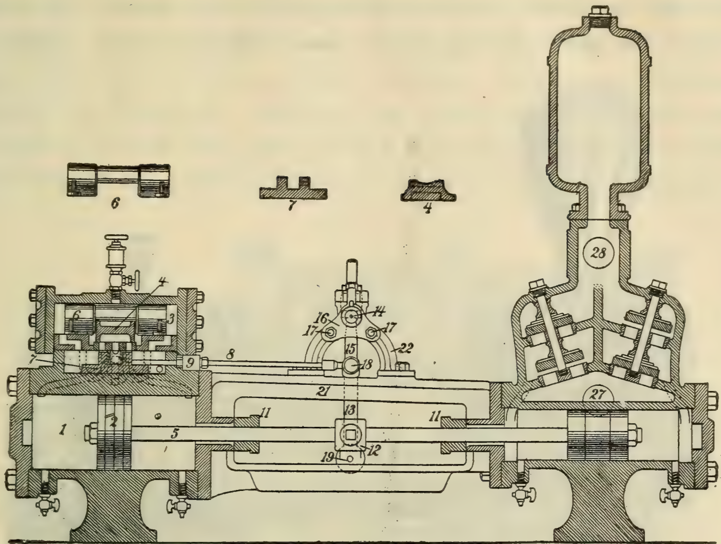


Fig. 286. Showing the steam valves properly set.

steam being supplied through both the main and auxiliary ports, but released through the main ports only.

BLAKE STEAM PUMP.

Description of the Blake steam pump. — The Blake steam pump is absolutely positive in its action; that is to say, the operation at the slowest speed under any pressure, is perfectly continuous, and the pump is never liable to stop as the main valve passes its center, if the pump is in good order. An ingenious and simple arrangement is used in the Blake pump to overcome the "dead center," as will be seen from the engraving, Fig. 288.

Operation of the Blake steam pump. — The main or pump driving piston *A* could not be made to work slowly were the main valve to derive its movement solely from this piston; for

when this valve had reached the center of its stroke, in which position the ports leading to the main cylinder would be closed,

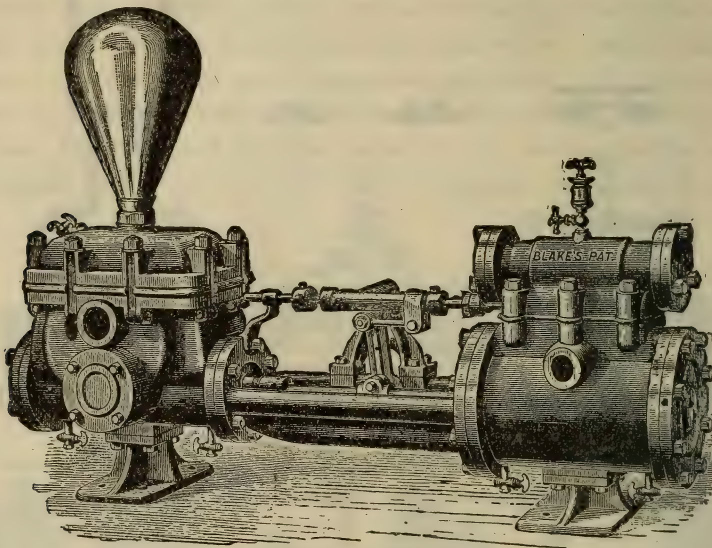


Fig. 287. The Blake steam pump.

no steam could enter the cylinder to act on said piston, consequently, the latter would come to rest, since its momentum would be insufficient to keep it in motion, and the main valve would remain in its central position or "dead center." To shift this valve from its central position and admit steam in front of the main piston (whereby the motion of the piston is reversed and its action continued), some agent independent of the main piston must be used. In the Blake pump, this independent agent is the supplemental or valve-driving piston *B*. The main valve, which controls the admission of steam to, and the escape of steam from, the main cylinder, is divided into two parts, one of which, *C*, slides upon a seat on the main cylinder, and, at the same time, affords a seat for the other part,

D, which slides upon the upper face of C. As shown in the engraving, D is at the left-hand end of its stroke, and C at the opposite, or right-hand end of its stroke. Steam from the steam-chest J is, therefore, entering the right-hand end of the main cylinder through the ports E and H, and the exhaust is escaping through the ports H^1 and E^1, K and M, which causes the

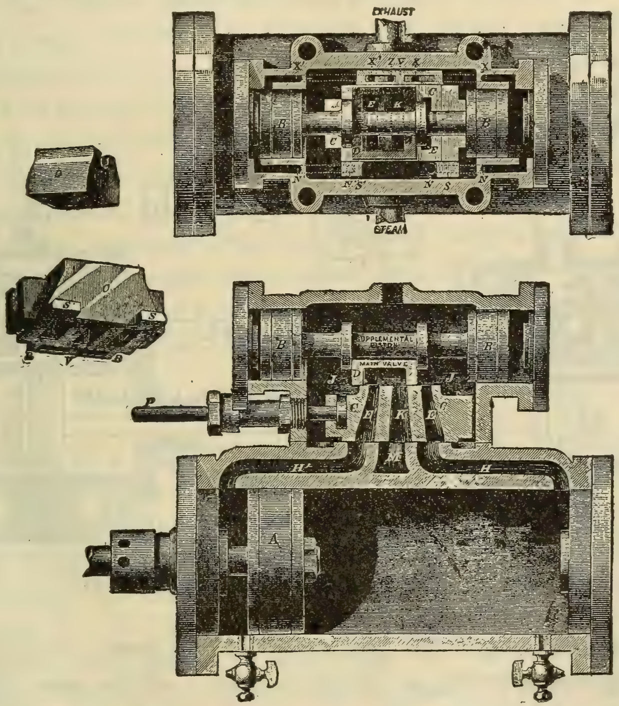


Fig. 288. Sectional views of steam cylinder, valves, etc., of the Blake steam pump.

main piston A to move from right to left. When this piston has nearly reached the left-hand end of its cylinder the valve motion

(not shown) moves the valve-rod P, and this causes C, together with its supplemental valve R and S S^1 (which form, with C, one casting) to be moved from right to left. This movement causes steam to be admitted to the left-hand end of the supplemental cylinder, whereby its piston B will be forced toward the right, carrying D with it to the opposite or right-hand end of its stroke; for the movement of S closes N (the steam port leading to the

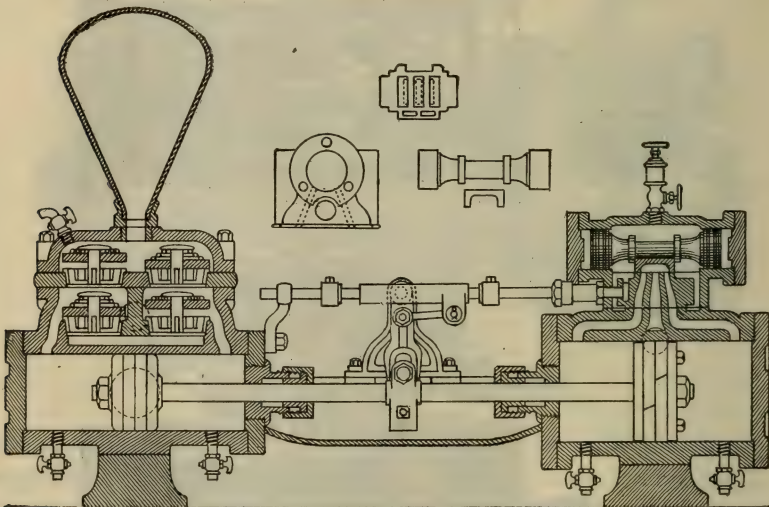


Fig. 289. Showing the valves properly set.

right-hand end), and the movement of S^1 opens N^1 (the port leading to the opposite, or left-hand end). At the same time the movement of O opens the right-hand end of the cylinder to the exhaust through the exhaust ports X and Z. The ports C and D now have positions opposite to those shown in the engravings, and steam is, therefore, entering the main cylinder through the ports E^1 and H^1, and escaping through the ports H, E, K and M, which will cause the main piston A to move in the op-

posite direction, or from left to right, and operations similar to those already described will follow, when the piston approaches the right-hand end of its cylinder. By this simple arrangement the pump is rendered positive in its action; that is, it will instantly start and continue working the moment steam is admitted to the steam chest. The main piston *A* cannot strike the head of the cylinder, for the main valve has a lead; or, in other words, steam is always admitted in front of said piston just before it reaches either end of its cylinder, even should the supplemental piston *B* be tardy in its action and remain with *D* at that end, toward which the piston *A* is moving; for *C* would be moved far enough to open the steam port leading to the main cylinder, since the possible travel of *C* is greater than that of *D*. The supplemental piston *B* cannot strike the heads of its cylinders, for in its alternate passage beyond the exhaust ports *X* and *X*, it cushions on the vapor intrapped in the ends of this cylinder.

MISCELLANEOUS PUMP QUESTIONS.

Q. What is a pump? A. It is hard to get a definition that will cover the whole ground. A pump may be said to be a mechanical contrivance for raising or transferring fluids; and as a general thing consists of a moving piece working in a cylinder or other cavity; the device having valves for admitting or retaining the fluids.

Q. What two classes of operations are included in the term "raising" fluids? A. They may be raised by drafting or suction, from their level to that of the pump; they may be raised from the level of the pump to a higher level.

Q. Do pumps always "raise" by either method, from one level to a higher one, the liquid which they transfer? A. No; in many cases the liquid flows by gravity to the pump; and in some it is delivered at a lower level than that at which it is received.

Q. Where a pump is not used for raising a liquid to a higher level, for what is it generally used? A. To increase or decrease its pressure.

Q. What classes of Liquids are handled by pumps? A. Air, ammonia, lighting gas, oxygen, etc.

Q. Name some liquids which are handled by pumps? A. Water, brine, beer, tan liquor, molasses, acids and oils.

Q. Where it is not specified whether a pump is for gas or for liquid, which is generally understood? A. Liquid.

Q. What gas is most frequently pumped? A. Air.

Q. What liquid is generally understood if none other is specified for a pump? A. Water.

Q. Can pumps handle hot and cold liquids? A. Yes; though cold are easier handled than hot.

Q. What is the difference between a fluid and a liquid? A. Every liquid is a fluid; every fluid is not a liquid. Air is a fluid; water is both a fluid and a liquid. Every liquid can be poured from one vessel to another.

SUCTION.

Q. What causes the water to rise in a pump by so-called suction? A. The unbalanced pressure of the air upon the surface of the liquid below the pump, forces the water up into the suction pipe when the piston is withdrawn from the liquid.

Q. How much is the pressure of the atmosphere? A. At the sea level about 14.7 lbs. per square inch, or 2116.8 lbs. per square foot.

Q. In what direction is this pressure exerted? A. In every direction equally.

Q. What tends to prevent the water from being lifted? A. The force of gravity, which is the result of the attraction of the earth.

Q. In what direction does the force of gravity act? A. In radial lines towards the center of the earth.

Q. With what force does this gravity act? A. That depends upon the substance upon which it is acting.

Q. Why do you refer to the level of the sea in speaking of the pressure of the air and the weight of water? A. Because the air pressure becomes less as, in rising above the sea level, we recede from the center of the earth, and the weight of a given quantity of water or any other substance becomes less than it is at the level of the sea, as we approach to or recede from the center of the earth.

Q. How is it that the weight of any substance becomes less if you go either above or below the sea level? A. The farther you go from the earth, the less its attraction and the less a given body will weigh upon a spring balance. The farther down into the earth you go, the nearer you get to the center of the earth, at which, there being attraction upon all sides, any body would weigh nothing. Going from the surface of the earth towards its center, then, a body weighs less and less upon a spring balance.

Q. Why do you specify a spring balance? A. Because in weighing by counterpoise, both the body to be weighed and the counterpoise by which it is weighed, would change their weights in the same proportion, as the position with regard to the center of the earth was changed.

Q. What are the causes which principally prevent pumps from lifting up to the normal maximum? A. Friction; leakage of air into the suction, chokes in the suction pipe.

Q. Can a liquid be "drafted" without the expenditure of work? A. No; in drafting a liquid to the full height to which it can be drafted, at least as much power must be expended as would lift the same weight of liquid that height by any mechanical means; only the amounts of friction being different.

Q. Then what advantage is there in having a pump draft its

water to the full possible height, over having it force the water the full height? A. Convenience in having the pump higher up.

Q. Can a pump throw water higher or farther, with a given expenditure of power, where it flows in, than where it must draft its water? A. Yes; on the same principle that it can throw farther or force harder when the water is forced to its suction side than where it merely flows in.

Q. What is the use of the suction chamber? A. To enable the pump barrel to fill where the speed is high; to prevent pounding, when the pump reverses.

Q. Upon what does the lifting capacity of a pump depend? A. When the pump is in good order its lifting capacity depends mainly upon the proportion of clearance in the cylinder and valve chamber to the displacement of the piston and plunger.

Q. Which will lift farther, an ordinary piston pattern pump or a plunger pump? And why? A. Other things being as nearly equal as they can be made between these two pumps, the piston pump will lift the farther of the two, because the plunger pump has the most clearance.

Q. What is the advantage of the suction chamber? A. To assist the pump in drafting, especially at high speed.

Q. What is the advantage of the air chamber? A. To make the stream steady.

Q. What difficulty is sometimes met with in using an air chamber? A. Where the pressure is very great sometimes the air is absorbed by the water, and thus the cushion is detroyed.

FORCING.

Q. What will be the volume of the air in the air chamber of a force pump, when the pump is forcing against a head of 67.6 feet? A. It will be reduced to half its ordinary volume, because it will be at the pressure of two atmospheres.

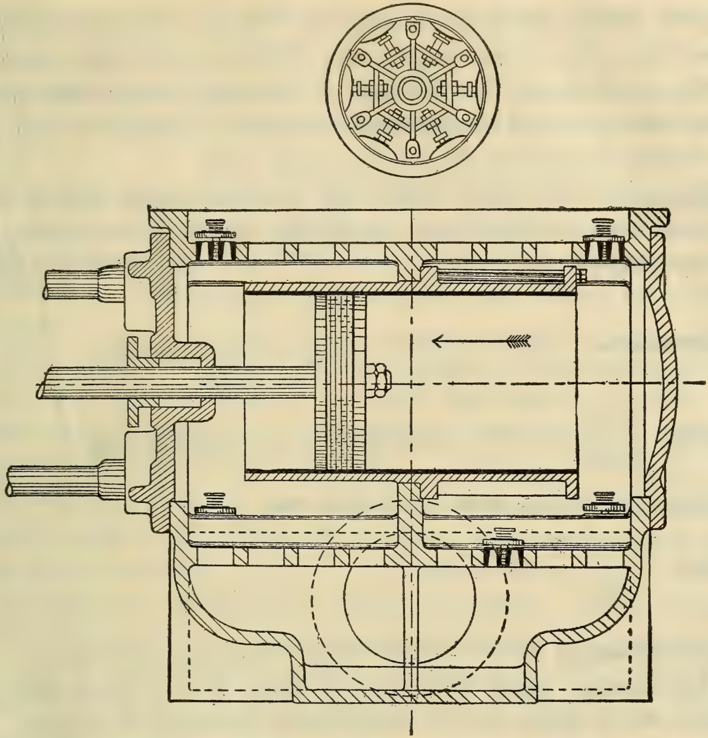


Fig. 290. Pump cylinder fitted with liner.

The above cut shows a pump with a removable cylinder or liner, and is packed with fibrous packing set out by adjustable set screws and nuts. This style of a pump is the best for small water-works or elevators, or where a pump is used where the water is muddy or sandy.

To find the horse power necessary to elevate water to a given height: Multiply the total weight of the water in pounds by the height in feet and divide the product by 33,000 (an allowance of 25 per cent should be added for water friction, and a further allowance of 25 per cent for loss in steam cylinder.)

The heights to which pumps will force water when running at

100 feet piston speed per minute, and the suction and discharge pipes being of moderate length, will be found by dividing the area of the steam piston by the area of the water piston, and multiplying the quotient by the steam pressure. Deduct 40 per cent for friction and divide the remainder by .434.

Example. — To what height will an 8-inch steam piston, with a 5-inch water piston, force water, the steam pressure being 80 lbs. by gauge? Ans. 283 ft. nearly.

Operation. — Area of steam piston = 50.26 sq. ins.
" " water " = 19.63 " "

Then, $\frac{50.26}{19.63} = 2.56$. And $2.56 \times 80 = 204.80$ lbs.

Then, 204.80 less 40% = 122.88 lbs.

And, $\frac{122.88}{.434} = 283 +$ feet.

An allowance must be made where long pipes are used.

The normal speed of pumps is taken at 100 piston feet per minute, which speed can be considerably increased if desired.

For feeding boilers, a speed of 25 to 50 lineal feet per minute is most desirable.

A gallon of water, U. S. Standard, weighs $8\frac{1}{3}$ lbs. and contains 231 cubic inches.

A cubic foot of water weighs 62.425 lbs. and contains 1,728 cubic inches, or $7\frac{1}{2}$ gallons.

Doubling the diameter of a pipe increases its capacity four times.

Friction of liquids in pipes increases as the square of the velocity.

To find the area of a piston, square the diameter and multiply by .7854.

Boilers require, for each nominal horse-power, about one cubic foot of feed water per hour.

In calculating horse power of tubular or flue boilers, consider 15 square feet of heating surface equivalent to one nominal horse-power.

To find the pressure in pounds per square inch of a column of water, multiply the height of a column in feet by .434. Approximately, we say that every foot of elevation is equal to one-half lb. pressure per square inch; this allows for ordinary friction.

The area of the steam piston, multiplied by the steam pressure, gives the total amount of pressure that can be exerted. The area of the water piston, multiplied by the pressure of water per square inch, gives the resistance. A margin must be made between the power and the resistance to move the pistons at the required speed — say from 20 to 40 per cent, according to speed and other conditions.

To find the capacity of a cylinder in gallons: Multiplying the area in inches by the length of stroke in inches will give the total number of cubic inches; divide this amount by 231 (which is the cubical contents of a gallon of water) and quotient is the capacity in gallons.

To find quantity of water elevated in one minute running at 100 feet of piston speed per minute: Square the diameter of water cylinder in inches and multiply by 4.

Example: Capacity of a five-inch cylinder is desired. The square of the diameter (5 inches) is 25, which, multiplied by 4, gives 100, which is gallons per minute, approximately.

Q. What is the reason that a steam pump of the horizontal double acting type should throw an intermitting stream under pressure, like the stream from milking a cow, only not quite so bad as that? Have tried valves of different sizes, with different amount of rise, springs or valves of different tension, different

kinds of packing in water piston, and different sized water ports or passages, without any apparent difference. A. Steam pumps of the horizontal double-acting type are not alone in throwing an intermitting stream. The same thing shows up in vertical single-acting pumps; but all horizontal double-acting pumps do not so behave. The steam fire engine shows that no type of pump is exempt from "squirting."

Q. How may this squirting be lessened? A. By increasing the suction valve area; by giving more suction chamber and more air chamber.

* * * * * * * * *

Q. What is a sinking pump? A. One which can be raised and lowered conveniently, for pumping out drowned mines, etc.

Q. Into what main general classes may reciprocating cylinder pumps be divided? A. Into single acting and double acting.

Q. What is a single acting reciprocating pump? A. One in which each reciprocation or single stroke in one direction causes one influx of fluid, and each reciprocation or single stroke in the opposite direction causes one discharge of fluid. In other words, the pump, as regards its action, is single ended.

Q. What is a double acting reciprocating pump? A. One in which each end acts alternately for suction and discharge. Reciprocation of the piston in one direction causes an influx of fluid into one end of the pump from the source, and a discharge of fluid at the opposite end; on the return stroke the former suction end becomes the discharge end. In other words, the pump is double ended in its action; or is "double-acting."

Q. What is the special advantage of having double-acting pump cylinders? A. The column of water is kept in motion more constantly, and hence there is less jar; smaller pipes may be used.

* * * * * * * * * * *

Q. How may those pumps which are driven by steam against a

steam piston be divided? A. Into those which have a fly wheel and those which have no fly wheel.

Q. Into what classes may those pumps which are driven by steam, without a fly wheel, be divided? A. Into direct acting and duplex.

Q. What is the advantage of a fly wheel steam pump? A. Steadiness of action; the capability of using the steam expansively.

Q. What are the disadvantages of fly wheel pumps? A. Great weight; inability to run them very slowly without gearing down from the fly wheel shaft, as the wheel must run comparatively rapidly.

Q. What is a direct-acting steam pump? A. One in which there is no rotary motion, the piston being reversed by an impulse derived from itself at or near the end of each stroke. There is but one steam cylinder for one water cylinder; the valve motion of the steam cylinder being controlled by the action of the steam in that cylinder.

HOW TO SET THE STEAM VALVES ON A DUPLEX PUMP.

The steam valves on duplex pumps generally have no outside lap, consequently, in their central position, they just cover the steam ports leading to the opposite ends of cylinder.

By lost motion is meant, the distance a valve-rod travels before moving the valve; if the steam-chest cover is off the amount of lost motion is shown by the distance the valve can be moved back and forth before coming in contact with the valve-rod nut. The object of lost motion is to allow one pump to almost complete its stroke before moving the valve of its fellow engine. As the steam piston is nearing the end of its stroke, it moves the valve of its fellow engine, admitting steam and starting its fellow engine as it lays down its own work; in other words,

the other picks it up. The amount of lost motion required is enough to allow each piston to complete its stroke; in other words, if there was no lost motion, as the pistons would pass the center of their travel, they would move the valve of their fellow engine, and the result would be a very short stroke.

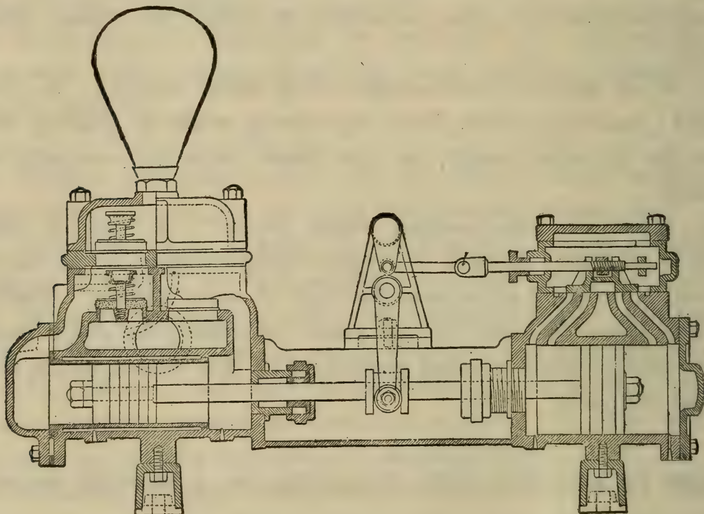


Fig. 291. Showing the steam valves properly set.

To set the steam valves, move the steam piston towards the steam cylinder head until it comes in contact with the head; mark with a scribe on the piston-rod at the face of the stuffing-box follower on steam end; then move the piston to its contact stroke on the opposite end and make another mark on the piston-rod, exactly half way between the face of the stuffing-box follower on the steam end, and the first mark. Then move the piston back until the middle mark is at the face of piston-rod stuffing-box follower on the pump end. This operation brings the piston exactly in the middle of the stroke. Then take off the steam

chest cover, place the slide-valve in the center, exactly over the steam ports. Place the slide-valve nut in exact center between the jaws of the slide-valve, screw the valve-rod through the nut until the eye on the valve-rod head comes in line with the eye of the valve-rod link ; slip the valve-rod head pin through head and the valve is set. Repeat the same operation on the other side of the pump. Where a pump is fitted with four hexagon valve-rod nuts, two either end of the slide-valve, instead of one nut in the center of the valve, set and lock these hexagon nuts at equal distances from the outer end of the slide-valve jaws, allowing a little lost motion, varying from $\frac{1}{2}''$ on high-pressure pumps, to, say, $\frac{1}{4}''$ on low service pumps, on each side of valve ; if the steam piston hits the head, take up some of the lost motion ; if the steam piston should not make a full stroke, give more lost motion.

PROPER MANNER OF ARRANGING PIPE CONNECTIONS.

For the purpose of showing good arrangement, the following cut is presented, Fig. 292.

On long lifts it is necessary to provide the suction pipe S with a foot-valve F. By the use of a foot-valve, the pipe and cylinders are constantly kept charged with water, allowing the pump to start without having to free itself and the suction pipe of air. In case of a long lift, the vacuum chamber V is also essential. This may be readily constructed by using a tee in place of the elbow E, extending the suction pipe and placing a cap upon the top. In order to keep the water back when the pump is being examined or repaired, a gate valve should be placed in the delivery pipe. It sometimes happens that, either purposely or through a leak in the foot-valve, the suction chamber becomes empty. For the purpose of charging the suction pipe and cylinder a "charging pipe" P is placed *outside* the check valve, connecting the delivery pipe D with the suction. In order that

the pump, in starting, may free itself of air, a check valve C and a "starting pipe" A should be provided. This pipe may be

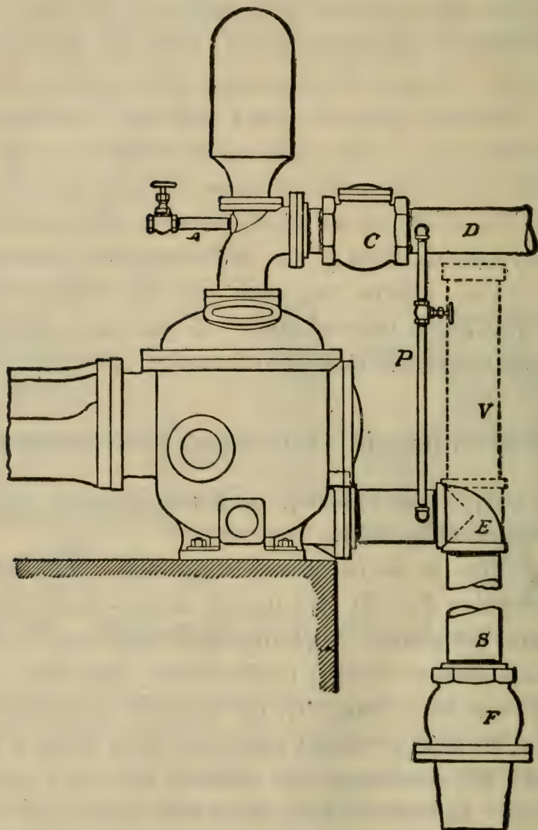


Fig. 292. Proper arrangement of pipe connections.

led to any convenient place of discharge. After the pump has started, the valve in the starting pipe should be closed gradually. Faulty connections are generally the cause of the improper action

of a pump. Great care should, therefore, be taken to have everything right before starting. A very small leak in the suction will cause a pump to work badly.

Q. What is the peculiarity of the duplex type? A. There are two steam cylinders and two water cylinders; the piston of one of these cylinders works the valve of the other cylinder, and *vice versa*. Neither half can work alone. This name is entirely arbitrary.

Q. What would you call a pumping machine in which there are two steam cylinders, each operating a water cylinder in line with it; each half being a perfect pumping machine independent of the other side? A. A "double" pump.

Q. Can a direct acting steam pump use steam expansively? A. Not to any extent; in fact, there would be danger of sticking upon the centers in most cases, if there was lap and expansion.

Q. What is the reason that a single cylinder engine cannot well reverse itself without a fly wheel, by means of the ordinary single *D* valve? A. Because when the valve was at mid-travel, both ports of the valve seat would be closed by the valve faces, and neither exhaust nor admission take place.

Q. What means are employed in a direct acting steam pump to move the valve? A. A small supplementary piston is used; this supplementary piston being actuated by the main piston in any one of several different ways.

Q. What are the principal ways of working the supplementary piston from the main piston? A. (1) The main piston strikes the tappet of a small valve, which opens an exhaust passage in one end of the cylinder, containing a supplementary piston, and having live steam pressing upon both ends of the supplementary piston; (2) by the main piston striking a rod passing through the cylinder head, and moving a lever, which controls the motion of the part of the main valve to which is attached the valves which moves the supplementary piston; (3) the main piston rod carries a tappet arm, which twists the stem of the supplementary piston,

thus uncovering ports which cause its motion; (4) a projection upon the main piston rod engages the stem and operates the valve which moves the supplementary piston, but if that valve should not, by means of its steam passages, cause quick enough or sure enough motion of the supplementary piston, a lug upon this stem moves the supplementary piston.

Q. In the first of these four classes, what is the principal element in the valve motion? A. A difference in area between the eduction port of the supplemental piston and its induction port.

Q. What is the principal feature in the second class? A. A regular slide valve letting steam upon alternate ends of the supplemental piston.

Q. In the third class, what is the main feature? A. A twisting motion in the supplemental piston.

Q. In the fourth class, what is the principal feature? A. Movement of the supplemental piston by steam controlled by a slide valve, and by the mechanical action of the slide valve itself if its steam distribution is defective.

Q. What are the objections to most pumps of the direct acting type? A. The unbalanced condition of the auxiliary pistons in the exhaust side, causing a loss of steam when the parts are worn, the choking up of the small ports for the auxiliary pistons, by the gumming and caking of the oil therein.

Q. Can the ordinary direct acting steam pump use steam expansively? A. No.

Q. How may this be done? A. By compounding.

Q. What is to be taken into consideration in the use of compound steam pumps? A. That they are designed for a certain range of pressure — say from 80 to 120 pounds boiler pressure, and will do their best work between these pressures.

Q. Have all direct acting steam pumps intermittent valve motion? A. No; there are some which have continuous valve motion.

Q. In most direct-acting steam pumps, are the auxiliary piston heads made together or in separate pieces? A. Together.

Q. They are in contact with the steam in the chest? A. Yes.

Q. What should be said about the location of a pump? A. It should be as near the source of supply as is convenient.

Q. What may be said about convenience in repairs? A. The pump should have room left upon all sides; and upon both ends equal to its length, for the removal of the piston rods in case of repairs.

Q. If the floor is not strong enough, how may a good foundation be made? A. By digging two or three feet into the ground and building up the proper height with stone or brick laid in strong cement, with a cap stone.

Q. What may be said about suction pipes? A. They must be as large as possible; the longer they are the greater in diameter they should be; they should be as straight as possible, and as free from bends and valves; they must be air-tight; they must not be allowed to get obstructed by foreign substances.

Q. What may be said about the area of strainer holes? A. They should have an aggregate area about five times that of the suction pipe.

Q. Where are foot valves necessary? A. Upon long suctions or high lifts.

Q. Should two pumps take their suction from one pipe? A. It should be avoided, unless the pipe is very large; and in case both suctions should be arranged so that one of the pumps should not have to draft at right-angles to the flow of water going to the other pump.

Q. What arrangement should be made where it is necessary to have two pumps draft from one suction? A. There should be a Y connection.

Q. What is a good way to reduce the friction in suction pipes where there are many bends? A. To use bends of wrought-

iron pipe of as long a radius as possible, instead of cast-iron elbows.

Q. What may be said about the lower end of the suction pipe? A. It should generally have a strainer; and if the lift is over 12 to 15 feet, should have a foot valve.

Q. What is a good thing to do with the discharge pipe near the pump? A. To put a valve in it near the pump, to keep the water in the pipe when the water end is to be opened for inspection or repairs.

Q. What provision should be made for priming the pump? A. There should be a pipe with a stop valve in it connected from the discharge pipe beyond this check valve, or from some other source of supply, to the suction pipe, for the purpose of priming the pump.

Q. When the pump is in position for piping, what care should be taken? A. That the pipes are of proper length, so as not to bring any undue strain upon them in connecting them to the pump, as in that case they will be liable to give trouble by breaking or working the joints loose and leaking.

Q. Does any pipe have an effective diameter as great as its nominal diameter? A. No; because the sides retard the flow of the liquid; there is a neutral film of liquid which practically does not move.

Q. Upon what does the thickness of this film of liquid depend? A. Upon the viscosity (commonly miscalled the "thickness") of the liquid; upon the roughness, material and diameter of the pipe; the pressure, etc.

Q. When long lines of pipe are used, should the diameter of the pipe be the same all the way along, or should there by sections be decreasing diameter, as the distance from the pump increases? A. Most emphatically, the pipe diameter should remain constant clear out to the end.

TAKING CARE OF A PUMP.

Q. What can be said about taking care of a pump? A. In places where an inferior grade of labor is employed, oil and dirt are sometimes found covering the steam chest and pump to the depth of an inch in thickness; stuffing boxes are allowed to go leaky and get loose; the valve motion is never looked after; lost motion is never taken up, and the pump will be let run in a slipshod way for months, until some accident occurs. This will sometimes exist in places where the engine is well taken care of.

Q. Should not as good care be taken of a steam pump as of an engine? A. Yes. It is a steam engine, and the fact that it has generally but little adjustability, should not render it liable to lack of care.

Q. What is a very common thing for pump runners to do when anything happens? A. To condemn the pump at once without finding out the cause of the trouble.

Q. What is one reason of this? A. The man who understands an ordinary engine, will often become quite perplexed when he examines the steam end of a direct acting steam pump, because he does not comprehend the principal feature of its construction — that all direct acting steam pumps which have no fly wheels and cranks, must generally have an auxiliary piston in order to carry them over the "dead center." A direct acting steam pump is really a double engine; a plain, flat slide valve admitting steam to a small piston, which in turn operates the main valve, which gives steam by the usual arrangement to the main piston.

Q. What would save firemen and engineers much trouble with steam pumps? A. If they would take the trouble to examine their pumps carefully, and find out the way their valves were arranged and actuated.

Q. Upon what does the successful performance of a pump

depend, in great measure? A. Upon its proper selection from among the many patterns differing from each other in size, proportion and general arrangement.

Q. What may be said about the selection of pumps? A. Pumps are often selected improperly for their work. As an illustration, a man who wishes to use a circulating pump for a surface condenser, where the water pressure upon the pump cylinder will never exceed 5 to 10 pounds, will buy a pump intended for boiler feed work, and having its steam cylinder about three times the area of its pump cylinder.

Q. What will be the result in such a case? A. There will be little or no pressure in the steam cylinder when working on the condenser; and while there is pressure sufficient to move the main piston, there is not enough to operate the auxiliary piston with positiveness.

Q. In ordering a pump, or in asking estimates, what information should be given? A. In ordering a pump, it is to the interest of the purchaser to fully inform the maker or seller on the following questions: 1st. For what purpose is the pump to be used? What is the average steam pressure? 2d. What is the liquid to be pumped; and is it hot or cold, clear or gritty, fresh, salt, alkaline or acidulous? 3d. What is the maximum quantity to be pumped per minute or hour? 4th. To what height is the liquid to be lifted by suction, and what is the length of the suction pipe, and the number of elbows or bends? 5th. To what height is the liquid to be pumped, and what is the length of discharge pipe?

Q. How can an engineer familiarize himself with the direction of the auxiliary steam and exhaust passages? A. By means of a piece of wire.

Q. What is the special thing to look after in duplex pumps? A. That all packings are adjusted uniformly on both sides.

Q. What would be the result of having the packings different

upon the two sides of a duplex pump? A. The machinery would run unsteadily.

Q. If a pump works badly, what should be about the first thing to look at? A. The connections.

Q. When a pump is first connected, what should be done? A. It should be blown through to remove dirt; if it be of the class which will permit of removing the bonnets and blowing through, that should be done.

Q. What pump piston speed is recommended for continuous boiler feeding service? A. About 50 feet per minute.

Q. What may be said about the care and use of steam pumps of all kinds? A. It is important that the pump be properly and thoroughly lubricated; that all stuffing-box, piston and plunger packings be nicely adjusted; not so tight as to cause undue friction; nor so slack as to leak badly.

Q. In which end of a steam pumping machine is there most likely to be trouble? A. In the water end.

Q. If a pump slams and hammers in its water end, is it necessarily defective in its water cylinder? A. No; it may be that there is no suction chamber, or not enough; or sometimes it slams because the suction pipe is not large enough.

Q. What are very common defects in cheap grades of pumps? A. Too little valve area in the pump end; too great lift for the valves.

Q. What are the principal causes of pumps refusing to lift water from the source of supply? A. Among these may be mentioned leaky suction pipes, worn out pistons, plungers, packings or water valves; rotten gaskets on joints in piping or pump; and sometimes a failure to properly prime the pump as well as the suction pipe.

Q. What is one great cause of a pump refusing to lift water when first started? A. It often happens that a pump refuses to lift water while the full pressure against which it is expected to

work is resting upon the discharge valves, for the reason that the air within the pump chamber is not dislodged, but only compressed, by the motion of the plunger. It is well, therefore, to arrange for running without pressure until the air is expelled and water follows; this is done by placing a valve in the delivery pipe and providing a waste delivery, to be closed after the pump has caught water.

Q. Sometimes when starting, the water may not come for a long time; what is the best thing to do in this case? A. First, open the little air cock, which is generally located in the top of the pump, between the discharge valves and the air chamber, to let off any accumulation of air, which may there be confined under pressure. Very often, by relieving the pump of this air pressure, it will pick up its water by suction and operate promptly.

Q. What precaution must be taken in priming the pump? A. The air cock, which should be provided at the top of the pump, should be opened to allow the escape of the air from the suction pipe and from the pump, and then the valve in the priming pipe should be opened. The pump should then be started slowly, as it aids in more completely filling the pump cylinders, which otherwise, might not occur and the pump might fail to lift water.

Q. Is there any advantage in having air in the suction? A. Sometimes a small amount of air let into the suction will cause less jarring when the duty is very heavy.

Q. What may be said about pumping hot water? A. Where the hot water is very hot, it should gravitate to the pump, instead of an attempt being made to draft it.

Q. In the plunger pumps, what is about the only wearing part of the water end? A. The packing of the plunger stuffing-boxes.

Q. How can a pump be prevented from freezing? A. By having draining cocks and opening them when the pump is idle.

Q. What may be said about leather piston packing for water cylinders? A. For cold water, or sandy, gritty water, the leather packing has many points to commend it; it makes a tight piston, and one that is the least destructive to pump cylinders.

Q. What is the best way to handle the square packing mostly employed, which is composed of alternate layers of cotton and rubber? A. Cut the lengths a trifle short, then there will be room for the packing to swell and not cause too much friction. We have known pistons where this precaution has not been taken to be fastened so securely in the cylinder by the swelling of the dry packing, that full steam pressure could not move them.

Q. What is the remedy in such a case? A. Remove the follower, take out the different layers of packing and shorten their lengths.

Q. What is the reason that some soft waters corrode pipes so often? A. Because they contain a large proportion of oxygen.

Q. Will a pump with a 6″ water cylinder and a 6″ steam cylinder force water into a boiler, the discharge from water cylinder being 4″ diameter; boiler pressure, 80 lbs.? A. A pump with a 6″ water cylinder and 6″ steam cylinder will not force water into the boiler which supplies it, no matter what the steam pressure, nor what the size of discharge pipe. It will not move. The pressures would be equalized and there would be nothing to overcome friction of steam and water in pipes and cylinder. The foregoing case supposes that the water is to be lifted to the pump; or at least that there shall be no head; also, that there shall be no fall from pump to boiler. If there were sufficient head or fall to overcome all the various frictions, and no lift, the pump would apparently work; but really, the water piston would be dragging the steam piston along.

Q. How may acids be pumped? A. By what is known as blowing up; that is, by employing a pump to put pressure upon

the acid in a closed vessel, thereby forcing it through a pipe placed in the bottom of the vessel.

Q. In case any wearing part of a pump gets to cutting, what should be done? A. If it is not practicable to stop the pump nor to reduce its speed, the part which is getting damaged should be given very liberal oiling.

Q. What is the best oil for this purpose? A. That depends on the nature of the cutting surfaces, and on the pressure therein; the mineral oils are generally more cooling than others, although they have less body to resist squeezing.

CALCULATING THE BOILER FOR A STEAM PUMP.

The amount of work which a boiler has to do is very easy of determination. Given the largest number of gallons which a pump will be required to pump per minute, and the height in feet from the surface of the well from which the water is drawn, to the point of discharge, you can easily tell by multiplying by $8\frac{1}{3}$ — the weight in pounds of one gallon — the number of foot pounds of power consumed per minute in lifting the water, adding a certain percentage for friction of the machine and of water in the pipe, we have the total number of foot pounds consumed per minute, and this divided by 33,000 will be the horse power consumed.

The allowance for friction will vary with the style, size and condition of the pump, the size of the pipe, and, above all, the manner in which the pipe is connected up, the number of right angle turns, etc.

This may be arrived at in another way. A column of water 2.3 feet in height exerts a pressure of one pound. Allowing the .3 for friction, we can, by dividing the total left in feet by two, get at the pressure per square inch, which is being exerted against the water piston or plunger, and multiplying by the number of

square inches in that piston gives the total pressure against which the pump is working. This multiplied by the piston speed in feet minutes, and divided by 33,000, will give the lift in horse power. In this case, as in the other, the lift must be calculated from the surface of the supply, and not from the pump, when the pump is lifting its supply. If the water flows to the pump it must be calculated from the height of the water cylinder. An allowance of, say, 25 per cent, should be made above the horse power thus shown, in order to provide for contingencies, and to be on the safe side.

In selecting a boiler to do this work, it must be borne in mind that a boiler which is sold for a certain horse power, is supposed to be able to furnish that power in connection with a good steam engine, and they are not apt to be overrated. Now, the steam pump as usually built, does not approach in economy the ordinary steam engine, and, therefore, a boiler which will develop twenty-five horse power in connection with a good engine would be too small for a pump which was required to do the same amount of work. The evaporation of 30 pounds of water per hour from feed at 100 degrees Fahr. into steam of 70 lbs. pressure, has been adopted by several authorities as a horse power. Any good automatic cut-off will run on this amount of water, and if an estimate can be made of the comparative performance of the pump under consideration, a close approximation to the desired size of boiler can be made.

THE WORTHINGTON WATER METER.

The counter registers cubic feet; one foot being $7\frac{48}{100}$ gallons, United States standard. It is read in the same way as registers of gas meters. The following example and directions may be of use to those unacquainted with the method: If a pointer is between two figures, the smaller one must invariably be taken. Suppose the pointers of the dials to stand as in the engraving.

The reading is 6,874 cubic feet. From the dial marked ten we get the figure 4; from the next, marked hundred, the figure 7; from the next, marked thousand, the figure 8; from the next,

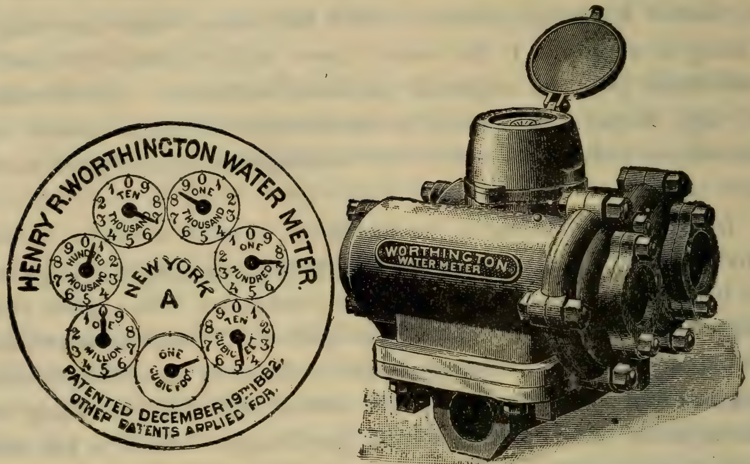


Fig. 293. Worthington water meter.

marked ten thousand, the figure 6. The next pointer being between ten and 1, indicates nothing. By subtracting the reading taken at one time, from that taken at the next, the consumption of water for the intermediate time is obtained.

TABLE OF PRESSURE DUE TO HEIGHT.

Feet head.	Equals pressure per sq. inch.	Feet head.	Equals pressure per sq. inch.	Feet head.	Equals pressure per sq. inch.	Feet head.	Equals pressure per sq. inch.	Feet head.	Equals pressure per sq. inch.	Feet head.	Equals pressure per sq. inch.	Feet head.	Equals pressure per sq. inch.
1	0.43	15	6.49	30	12.99	45	19.49	60	25.99	75	32.48	90	38.98
5	2.16	20	8.66	35	15.16	50	21.65	65	28 15	80	34 65	95	41.15
10	4.33	25	10.82	40	17.32	55	23.82	70	30.32	85	36.82	100	43.31

TABLE OF DECIMAL EQUIVALENTS OF 8ths, 16ths, 32ds AND 64ths OF AN INCH.

8ths.

$\frac{1}{8}$ = .125
$\frac{1}{4}$ = .25
$\frac{3}{8}$ = .375
$\frac{1}{2}$ = .50
$\frac{5}{8}$ = .625
$\frac{3}{4}$ = .75
$\frac{7}{8}$ = .875

16ths.

$\frac{1}{16}$ = .0625
$\frac{3}{16}$ = 1875
$\frac{5}{16}$ = .3125
$\frac{7}{16}$ = .4375
$\frac{9}{16}$ = .5625
$\frac{11}{16}$ = .6875
$\frac{13}{16}$ = .8125
$\frac{15}{16}$ = .9375

32ds.

$\frac{1}{32}$ = .03125
$\frac{3}{32}$ = .09375
$\frac{5}{32}$ = .15625
$\frac{7}{32}$ = .21875
$\frac{9}{32}$ = .28125
$\frac{11}{32}$ = .34375
$\frac{13}{32}$ = .40625
$\frac{15}{32}$ = .46875
$\frac{17}{32}$ = .53125
$\frac{19}{32}$ = .59375
$\frac{21}{32}$ = .65625
$\frac{23}{32}$ = .71875
$\frac{25}{32}$ = .78125
$\frac{27}{32}$ = .84375
$\frac{29}{32}$ = .90625
$\frac{31}{32}$ = .96875

64ths.

$\frac{1}{64}$ = .015625
$\frac{3}{64}$ = .046875
$\frac{5}{64}$ = .078125
$\frac{7}{64}$ = .109375
$\frac{9}{64}$ = .140625
$\frac{11}{64}$ = .171875
$\frac{13}{64}$ = .203125
$\frac{15}{64}$ = .234375
$\frac{17}{64}$ = .265625
$\frac{19}{64}$ = .296875
$\frac{21}{64}$ = .328125
$\frac{23}{64}$ = .359375
$\frac{25}{64}$ = .390625
$\frac{27}{64}$ = .421875
$\frac{29}{64}$ = .453125
$\frac{31}{64}$ = .484375
$\frac{33}{64}$ = .515625

64ths.

$\frac{35}{64}$ = .546875
$\frac{37}{64}$ = .578125
$\frac{39}{64}$ = .609375
$\frac{41}{64}$ = .640625
$\frac{43}{64}$ = .671875
$\frac{45}{64}$ = .703125
$\frac{47}{64}$ = .734375
$\frac{49}{64}$ = .765625
$\frac{51}{64}$ = .796875
$\frac{53}{64}$ = .828125
$\frac{55}{64}$ = .859375
$\frac{57}{64}$ = .890625
$\frac{59}{64}$ = .921875
$\frac{61}{64}$ = .953125
$\frac{63}{64}$ = .984375

LATENT HEAT OF LIQUIDS, UNDER A PRESSURE OF 30 INCHES OF MERCURY.

(TREATISE ON HEAT, BY THOMAS BOX.)

	Latent Heat in Units.	Increase of Temperature of Liquid, if Heat had not become Latent.	
Water..................	**966**	**966°**	Regnault.
Alcohol..................	457	735°	Ure.
Ether..................	313	473°	"
Oil of Turpentine.......	**184**	**390°**	"
Naphtha................	184	443°	"

The boiling point of different liquids varies; and the boiling point of a liquid varies with the pressure

CAPACITY OF TANKS IN U. S. GALLONS.

DIAMETERS.

	5 ft.	5½ ft.	6 ft.	6½ ft.	7 ft.	7½ ft.	8 ft.	8½ ft.	9 ft.	9½ ft.	10 ft.	11 ft.
5 ft.	734	888	1057	1241	1439	1652	1880	2122	2379	2768	2937	3554
5 ft. 6 in.	808	977	1163	1365	1583	1817	2068	2334	2617	3045	3231	3910
6 ft.	881	1066	1269	1489	1727	1983	2256	2546	2855	3322	3525	4265
6 ft. 6 in.	955	1155	1374	1613	1871	2148	2444	2759	3093	3599	3819	4621
7 ft.	1028	1244	1480	1737	2015	2313	2632	2971	3331	3875	4112	4976
7 ft. 6 in.	1100	1332	1586	1861	2159	2478	2820	3183	3569	4152	4406	5332
8 ft.	1175	1421	1692	1986	2303	2643	3007	3395	3907	4429	4700	5687
8 ft. 6 in.	1248	1510	1797	2110	2446	2809	3196	3607	4045	4706	4993	6042
9 ft.	1322	1599	1903	2234	2590	2974	3384	3820	4283	4983	5287	6398
9 ft. 6 in.	1395	1688	2008	2358	2734	3139	3572	4032	4521	5259	5581	6753
10 ft.	1468	1766	2114	2482	2878	3304	3760	4244	4758	5536	5874	7108

TANKS.

Length of Stave......ft.	6	5	8	6	10	7	8	10	12
Diam. of Bottom.....ft.	6	7	6	8	6	8	9	8	8
Number of Hoops....ft.	5	4	7	5	8	6	7	8	9
Capacity in Gallons.....	1017	1197	1480	1890	1890	2268	3339	3370	4126
Length of Stave......ft.	8	10	12	10	14	12	10	14	
Diam. of Bottom.....ft.	10	9	9	10	9	10	12	10	
Number of Hoops....ft.	7	8	9	8	9	9	8	10	
Capacity in Gallons.....	4126	4284	5229	5292	6174	6457	7623	7623	

CAPACITY OF SQUARE CISTERNS IN U. S. GALS.

	5×5	5×6	5×7	5×8	5×9	5×10	6×6	6×7	6×8	6×9	6×10
5 ft..	935	1122	1309	1496	1683	1870	1346	1571	1795	2020	2244
5½ ft..	1028	1234	1440	1645	1851	2057	1481	1728	1975	2221	2469
6 ft..	1122	1346	1571	1795	2019	2244	1615	1885	2154	2423	2693
6½ ft..	1215	1459	1702	1945	2188	2431	1750	2042	2334	2625	2917
7 ft..	1309	1571	1833	2094	2356	2618	1884	2199	2513	2827	3142
7½ ft..	1403	1683	1963	2244	2524	2800	2019	2356	2693	3029	3366
8 ft..	1496	1795	2094	2393	2693	2992	2154	2513	2872	3231	3592
8½ ft..	1589	1907	2225	2543	2861	3179	2288	2670	3052	3433	3816
9 ft..	1683	2020	2356	2693	3029	3366	2423	2827	3231	3635	4041
9½ ft..	1776	2132	2487	2842	3197	3553	2558	2984	3412	3837	4265
10 ft..	1870	2244	2618	2992	3366	3470	2692	3142	3591	4039	4489

	6×11	6×12	7×7	7×8	7×9	7×10	7×11	7×12	8×8	8×9
5 ft..	2468	2693	1832	2094	2356	2618	2880	3142	2394	2693
5½ ft..	2715	2962	2016	2304	2592	2880	3168	3456	2633	2962
6 ft..	2962	3231	2199	2513	2827	3142	3456	3770	2872	3231
6½ ft..	3209	3500	2382	2722	3063	3403	3744	4084	3112	3500
7 ft..	3455	3770	2565	2932	3298	3665	4032	4398	3351	3770
7½ ft..	3702	4039	2748	3141	3534	3927	4320	4712	3590	4039
8 ft..	3949	4308	2932	3351	3770	4189	4608	5026	3830	4308
8½ ft..	4196	4577	3115	3560	4005	4451	4896	5340	4069	4578
9 ft..	4443	4847	3298	3769	4341	4712	5184	5655	4308	4847
9½ ft..	4689	5116	3481	3979	4576	4974	5472	5969	4548	5116
10 ft..	4936	5386	3664	4188	4712	5236	5760	6283	4788	5386

WEIGHT OF WATER.

1 cubic inch	.03617	pound.
12 cubic inches	.434	pound.
1 cubic foot (salt)	64.3	pounds.
1 cubic foot (fresh)	62.425	pounds.
1 cubic foot	7.48	U. S. Gallons.

NOTE. — The center of pressure of a body of water is at two-thirds the depth from the surface.

To find the pressure in pounds per square inch of a column of water, multiply the height of the column in feet by .434. Every foot elevation is called (approximately) equal to one-half pound pressure per square inch.

SHOWING U. S. GALLONS IN GIVEN NUMBER OF CUBIC FEET.

Cubic Feet.	Gallons.	Cubic Feet.	Gallons.	Cubic Feet.	Gallons.
0.1	0.75	50	374.0	9,000	67,324.6
0.2	1.50	60	448.8	10,000	74,805.2
0.3	2.24	70	523.6	20,000	149,610.4
0.4	2.99	80	598.4	30,000	224,415.6
0.5	3.74	90	673.2	40,000	299,220.7
0.6	4.49	100	748.0	50,000	374,025.9
0.7	5.24	200	1,496.1	60,000	448,831.1
0.8	5.98	300	2,244.1	70,000	523,636.3
0.9	6.73	400	2,992.2	80,000	598,441.5
1	7.48	500	3,740.2	90,000	673,246.7
2	14.9	600	4,488.3	100,000	748,051.9
3	22.4	700	5,236.3	200,000	1,496,103.8
4	29.9	800	5,984.4	300,000	2,244,155.7
5	37.4	900	6,732.4	400,000	2,992,207.6
6	44.9	1,000	7,480.0	500,000	3,740,259.5
7	52.4	2,000	14,961.0	600,000	4,488,311.4
8	59.8	3,000	22,441.5	700,000	5,236,363.3
9	67.3	4,000	29,922.0	800,000	5,984,415.2
10	74.8	5,000	37,402.6	900,000	6,732,467.1
20	149.6	6,000	44,883.1	1,000,000	7,480,519.0
30	224.4	7,000	52,363.6		
40	299.2	8,000	59,844.1		

From the above any cubic feet reading can readily be converted into U. S. gallons, as follows:

How many gallons are represented by 53,928 cubic feet?

50,000	cubic	feet	=	374,025.9	gallons.
3,000	"	"	=	22,441.5	"
900	"	"	=	6,732.4	"
20	"	"	=	149.6	"
8	"	"	=	59.8	"
53,928	cubic	feet	=	403,409.2	gallons.

SHOWING COST OF WATER AT STATED RATES PER 1000 GALLONS.

Number of Cubic Feet.	COST PER 1000 GALLONS.							
	5 Cents.	6 Cents.	8 Cents.	10 Cents.	15 Cents.	20 Cents.	25 Cents.	30 Cents.
20	$0 007	$0.009	$0.012	$0.015	$0.021	$0.030	$0.037	$0.045
40	0.015	0.018	0.024	0.030	0.045	0.060	0.075	0.090
60	0.022	0.027	0.036	0.045	0.066	0.090	0.112	0.135
80	0.030	0.036	0.048	0.060	0.090	0.120	0.150	0.180
100	0.037	0.049	0.060	0.075	0.111	0.150	0.187	0.224
200	0.075	0.090	0.120	0.150	0.225	0.299	0.374	0.449
300	0.112	0.135	0.180	0.224	0.336	0.449	0.561	0.673
400	0.150	0.180	0.239	0.299	0.450	0.598	0.748	0.898
500	0.188	0.224	0.299	0.374	0.564	0.748	0.935	1.122
600	0.224	0.269	0.359	0.449	0.448	0.898	1.122	1.346
700	0.262	0.314	0.419	0.524	0.786	1.047	1.309	1.571
800	0.299	0.350	0.479	0.598	0.897	1.197	1.496	1.795
900	0.337	0.404	0.539	0.673	1.011	1 346	1.683	2.020
1,000	0.374	0.449	0.598	0.748	1.122	1.496	1.870	2.244
2,000	0.748	0.898	1.197	1.496	2.244	2.992	3.740	4.488
3,000	1.122	1.346	1.795	2.244	3.366	4.488	5.610	6.732
4,000	1.496	1.795	2.393	2.992	4.488	5.984	7.480	8.976
5,000	1.870	2.244	2.992	3.740	5.610	7.480	9.350	11.220
6,000	2.244	2.692	3.590	4.488	6.732	8.976	11.220	13.464
7,000	2.618	3.141	4.189	5.236	7.854	10.472	13.090	15.708
8,000	2.992	3.590	4.787	5.984	8.976	11.968	14.961	17.953
9,000	3.366	4.039	5.385	6.732	10.098	13.464	16.831	20.197
10,000	3.74	4.488	5.984	7.480	11.122	14.961	18.701	22.441
20,000	7.48	8.976	11.968	14.961	22.443	29.992	37.402	44.882
30,000	11.22	13.46	17.95	22.44	33.664	44.88	56.10	67.32
40,000	14.96	17.95	23.94	29.92	44.885	59.84	74.10	89.77
50,000	18.70	22.44	29.92	37.40	56.103	74.80	93.50	112.20
60,000	22.44	26.92	35.90	44.88	67.323	89.76	112.20	134.64
70,000	26.18	31.41	41.89	52.36	78.543	104.72	130.90	157.08
80,000	29.92	35.90	47.87	59.84	89.766	119.68	149.61	179.53
90,000	33.66	40.39	53.85	67.32	100.986	134 64	168.31	201.97
100,000	37.40	44.88	59.84	74.80	111.22	149.61	187.01	224.41
200,000	74.81	89.76	119.68	149.61	224.43	299.22	374.02	448.82
300,000	112.20	134.64	179.53	224.41	336.64	448.83	561.03	673.24
400,000	149.61	179.53	239.37	299.22	448.85	598.44	748.05	897.66
500,000	187.01	224.41	299.22	374.02	561.03	748.05	935.06	1122.07
600,000	224.41	269.29	359.06	448 83	673.23	897.66	1122.07	1346.49
700,000	261.81	314.18	418.90	523.63	785.43	1047.27	1309.08	1570.88
800,000	299.22	359.06	478.75	598.44	897.66	1196 88	1496.10	1795.32
900,000	336.62	403.94	538.59	673.24	1009.86	1346.49	1683.11	2019.73
1,000,000	374.02	448.83	598.44	748.05	1122.06	1498.10	1870.12	2244.15

LOSS BY FRICTION OF WATER IN PIPES.

This table shows the loss in pounds pressure per square inch for each 100 feet in length due to friction, when discharging the given quantities of water per minute.

Gals. per minute.	SIZES OF PIPES — INSIDE DIAMETER.														
	¾ Inch.	1 Inch.	1¼ Inch.	1½ Inch.	2 Inch.	2½ Inch.	3 Inch.	4 Inch.	6 Inch.	8 Inch.	10 Inch.	12 Inch.	14 Inch.	16 Inch.	18 Inch.
5	3.3	0.84	0.31	0.12											
10	13.0	3.16	1.05	0.47	0.12										
15	28.7	6.98	2.38	0.97											
20	50.4	12.3	4.07	1.66	0.42										
25	78.0	19.0	6.40	2.62		0.21	0.10								
30		27.5	9.15	3.75	0.91										
35		37.0	12.4	5.05											
40		48.0	16.1	6.52	1.60										
45			20.2	8.15											
50			24.9	10.0	2.44	0.81	0.35	0.09							
75			56.1	22.4	5.32	1.80	0.74								
100				39.0	9.46	3.20	1.31	0.33	0.05						
125					14.9	4.89	1.99								
150					21.2	7.0	2.85	0.69	0.10						
175					28.1	9.46	3.85								
200					37.5	12.47	5.02	1.22	0.17						
250						19.66	7.76	1.89	0.26	0.07	0.03	0.01			
300						28.06	11.2	2.66	0.37	0.09	0.04				
350							15.2	3.65	0.50	0.12	0.05	0.02			
400							19.5	4.73	0.65	0.16	0.06				
450							25.0	6.01	0.81	0.20	0.07	0.03			
500							30.8	7.43	0.96	0.25	0.09	0.04	0.017	0.009	0.005
750									2.21	0.53	0.18	0.08			
1000									3.88	0.94	0.32	0.13	0.062	0.036	0.020
1250										1.46	0.49	0.20			
1500										2.09	0.70	0.29	0.135	0.071	0.040
1750											0.95	0.38			
2000											1.23	0.49	0.234	0.123	0.071
2250												0.63			
2500												0.77	0.362	0.188	0.107
3000												1.11	0.515	0.267	0.150
3500													0.697	0.365	0.204
4000													0.910	0.472	0.263
4500														0.593	0.333
5000														0.730	0.408

SHOWING HOW WATER MAY BE WASTED.

GALLONS DISCHARGED PER HOUR THROUGH VARIOUS SIZED ORIFICES UNDER STATED PRESSURES.

Head in Feet.	Pounds pressure per square inch.	Diameters of Orifices in Inches and Fractions of an Inch.									
		¼ inch	⅜ inch	½ inch	⅝ inch	¾ inch	1 inch	1¼ inch	1½ inch	1¾ inch	2 inch
20	8.66	300	720	1260	1920	2760	4920	7380	11100	15120	19740
40	17.32	450	960	1800	2760	3960	6720	10920	15720	21360	27960
60	25.99	540	1200	2160	3480	4800	8580	13380	19200	26220	34260
80	34.65	620	1380	2460	3840	5580	9840	15480	22260	30300	39540
100	43.31	690	1560	2760	4320	6240	11040	17280	24900	33900	44280
120	51.98	780	1780	3000	4740	6840	12120	18960	27240	37440	48480
140	60.64	816	1860	3300	5100	7320	13020	20160	29460	39080	52320
150	64.97	840	1920	3420	5280	7620	13560	21180	30480	41460	54120
175	75.80	900	2040	3660	5700	8220	14640	22800	32880	44940	58560
200	86.63	960	2220	3900	6120	8760	15600	25020	35880	47880	62580
235	101.79	1080	2460	4320	8280	11160	17100	26760	38520	52260	68460

The pressure or head of water is taken at the orifice, no allowance being made for friction in the pipe. In practical calculations to determine the height which water can be thrown, the head consumed by the friction of the water in flowing from the source to the orifice must be considered.

IGNITION POINTS OF VARIOUS SUBSTANCES.

Phosphorus ignites at	150°	Fahr.
Sulphur " "	500°	"
Wood " "	800°	"
Coal " "	1000°	"
Lignite, in the form of dust, ignites at	150°	"
Cannel Coal, " " " "	200°	"
Coking Coal, " " " "	250°	"
Anthracite, " " " "	300°	"

PIPES AND TANKS.

CONTENTS IN CUBIC FEET AND IN U. S. GALLONS.

(FROM TRAUTWINE)

Of 231 cubic inches (or 7.4805 gallons to a cubic foot); and for one foot of length of the cylinder. For the contents for a greater diameter than any in the table take quantity opposite one-*half* said diameter, and multiply it by 4. Thus, the number of cubic feet in one foot length of a pipe 80 inches in diameter is equal to 8.728×4=34.912 cubic feet. So also with gallons and areas.

Diameter in inches.	Diameter in decimals of a foot.	For 1 foot in length.		Diameter in inches.	Diameter in decimals of a foot.	For 1 foot in length.		Diameter in inches.	Diameter in decimals of a foot.	For 1 foot in length.	
		Cubic feet, also area in square feet.	Gallons of 231 cubic inches.			Cubic feet, also area in square feet.	Gallons of 231 cubic inches.			Cubic feet, also area in square feet.	Gallons of 231 cubic inches.
¼	.0208	.0003	.0026	¾	.5625	.2485	1.859	19.	1.583	1 969	14.73
5-16	.0260	.0005	.0040	7.	.5833	.2673	1 999	½	1.625	2 074	15 52
⅜	.0313	.0008	.0057	¼	.6042	.2868	2.144	20.	1.666	2.182	16.32
7-16	.0365	.0010	.0078	½	.6250	.3068	2.295	½	1.708	3.292	17.15
½	.0417	.0014	.0102	¾	.6458	.3275	2.450	21.	1.750	2.405	17.99
9-16	.0469	.0017	.0129	8.	.6667	.3490	2.611	½	1.792	2.521	18.86
⅝	.0521	0021	.0159	¼	.6875	.3713	2 777	22.	1 833	2.640	19.75
11-16	.0573	.0026	.0193	½	.7083	.3940	2 948	½	1.875	2.761	20.65
¾	.0625	.0031	.0230	¾	.7292	.4175	3.125	23	1.917	2.885	22.58
13-16	.0677	.0031	.0270	9.	.7500	.4418	3 305	½	1.958	3 012	31.53
⅞	.0729	.0042	0312	¼	.7708	.4668	3.492	24.	2.000	3.142	23.50
15-16	.0781	.9048	0359	½	.7917	.4923	3 682	25.	2.083	3.409	25.50
1.	.0833	0055	.0408	¾	.8125	.5185	3.879	26.	2.166	3.687	27.58
¼	.1042	.0085	.0638	10.	.8333	.5455	4 081	27.	2.250	3.976	29.74
½	.1250	-0123	.0918	¼	8542	.5730	4.286	28.	2 333	4.276	31.99
¾	.1458	.0168	.1250	½	.8750	.6013	4.498	29.	2 416	4 587	34.31
2.	.1667	.0218	.1632	¾	.8958	.6303	4.714	30.	2.500	4.909	36.72
¼	.1875	.0276	.2066	11.	.9167	.6600	4.937	31.	2 583	5.241	39.21
½	.2083	.0341	.2550	¼	.9375	.6903	5.163	32.	2.666	5.585	41.78
¾	.2292	.0413	.3085	½	.9583	.7213	5 395	33.	2.750	5.940	44.43
3.	.2500	.0491	.3673	¾	.9792	.7530	5.633	34.	2.833	6.305	47.17
¼	.2708	.0576	.4310	12.	1 Foot.	.7854	5.876	35.	2.916	6.681	49.98
½	.2917	.0668	.4998	½	1.042	.8523	6.375	36.	3.000	7.069	52.88
¾	.3125	.0767	.5738	13.	1.083	.9218	6.895	37.	3.083	7.468	55.86
4.	.3333	.0873	.6528	½	1.125	.9940	7.435	38	3.166	7.876	58.92
¼	.3542	.0985	.7370	14.	1.167	1.069	7.997	39.	3 250	8 296	62.06
½	.3750	.1105	.8263	½	1.208	1.147	8.578	40.	3.333	8.728	65.29
¾	.3958	.1231	.9205	15.	1.250	1.227	9.180	41.	3.416	9.168	68.58
5.	.4167	.1364	1.020	½	1.292	1.310	9.801	42.	3.500	9.620	71.96
¼	.4375	.1503	1.124	16.	1.333	1.396	10.44	43	3.583	10.084	75.43
½	.4583	1650	1.234	½	1.375	1.485	11.11	44.	3 666	10.560	79 00
¾	.4792	.1803	1.349	17.	1.417	1.576	11.79	45.	3.750	11.044	82 62
6.	.5000	.1963	1.469	½	1.458	1.670	12.50	46.	3.833	11.540	86.32
¼	.5208	.2130	1.594	18.	1.500	1.767	13.22	47.	3.916	12.048	90.12
½	.5417	.2305	1.724	½	1.542	1.867	13.97	48.	4.000	12.566	94.02

CHAPTER XX.

THE INJECTOR AND INSPIRATOR.

The energy of motion of a body is well known to be the product of its mass by the half square of its velocity; hence, it is possible to communicate to a body of little weight a large amount of energy by moving it fast enough, and in fact, the energy of motion would only be limited by the speed which can be given the body. In this way a small weight of steam flowing from an orifice into a properly shaped jet of water is condensed, while the velocity of the steam is greater than if flowing into air; the energy thus communicated is made sufficiently great by increasing the weight of steam, which can be done by increasing the area of the steam way, until we find such jet pumps adapted to many purposes. There are, however, two which are of interest to us in this connection, the well-known injector and inspirator, with the large number of lifting and non-lifting varieties, all differing in details as to form of nozzles, area of passages, distances between nozzles, and that class of instruments in which, after a certain energy and velocity have been reached, the operation is repeated. These might be called "consecutive" instruments. The illustrations in this book show some of the simplest and adjustable kinds. Within a few years the principle of increase of energy by increase of mass or velocity has been applied by increasing the mass of steam used until we find that not only can a few pounds weight of steam put into a boiler, a good many more pounds of water at a much higher temperature than it had, but that in a non-condensing engine it is possible, by using the exhaust in part, to put into the boiler at a much higher pressure

and temperature, a weight of water, which is still greater than that of the steam moving it.

When the injector first made its appearance it was, by many, considered as almost a paradox, especially by those who looked at the question as one of hydrostatics only. That steam from a boiler could put water back into it at the same pressure, and overcome the friction of the passages without the aid that a steam pump had of a difference of piston areas, was to them a puzzle. The use of exhaust steam at atmospheric pressure for the purpose of putting water into a boiler at a pressure of 150 lbs. per square inch, would be to such minds utterly incomprehensible. The use of an injector and inspirator, has this to recommend them, that the feed-water cannot be introduced into the boiler cold or nearly so, but must be warmed by contact with the steam, and the value of this has been already shown. In small boilers where no heater is used, an exhaust injector is better than a pump, and so is an ordinary injector; but the former includes in itself an exhaust heater, saving a portion of heat from the exhaust, besides taking the power as heat also; while, with the common injector, the heat for power and raising temperature are both derived from the live steam in the boiler. The latter portion of heat is, of course, directly returned to the boiler without loss, but that for power is necessarily expended. As to the amount of power used by pump and injector compared with each other, it would seem that the pump is most efficient. There have been many comparative trials of pump and injector, but the results have usually been unsatisfactory from contained discrepancies.

RANGE OF THE INSPIRATOR AND INJECTOR.

The steam pressure at which an injector will start and the highest steam pressure at which it will work constitute what is termed the "range" of an injector, and the inspirator varies with the vertical lift and the temperature of the feed water.

It must also be borne in mind that the same style of construction in an injector and inspirator, while it confines them to about a specific range between its lowest starting and highest working points, permits of variation as to what the lowest starting point shall be. A style of construction, which gives a range (on say a 2-foot lift) of 25 lbs. to 155 lbs. would permit of a range of 35 lbs. to 165 lbs. (in fact, to a little higher than 165 lbs.). Different manufacturers, therefore, vary as to the starting point in their standard machines — aiming to cover the range which they deem most desirable. Nearly all have adopted about 25 lbs. on a 2-foot lift, as lowest starting point.

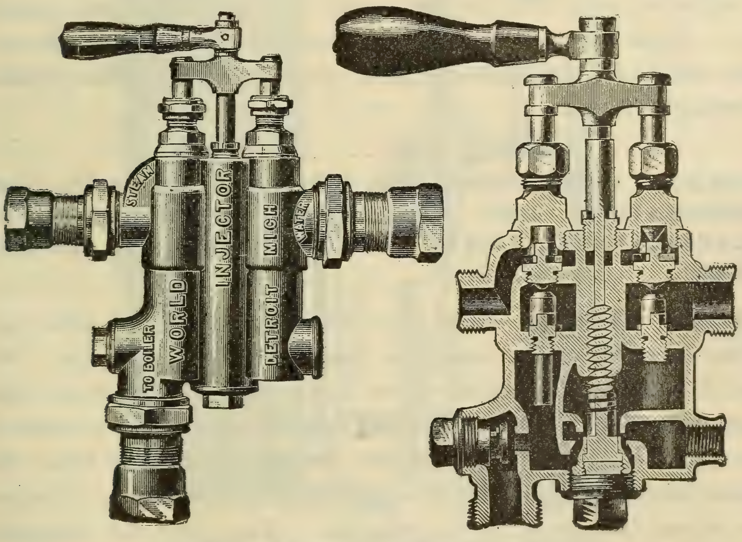


Fig. 294. The World injector.

POSITIVE OR DOUBLE TUBE INJECTORS.

As before stated, this class of injector is provided with two sets of tubes or jets, one set adapted to lift the water and deliver it to

the second set, which forces the water into the boiler. By this arrangement, it is apparent that inasmuch as the lifting jets supply a proportionate amount of water with varying steam pressures, a wider range is obtainable than with an automatic injector. In the following cases, it is better to use the double tube injectors: —

1. Where the feed water is of too high a temperature to be handled by the automatic injectors.

2. When a great range of steam variation is accompanied by the condition of a long lift.

The World injector is one of the simplest boiler feeders of the double tube type of injectors. It is entirely self contained. It is supplied even with its own check valve and operated entirely by a single lever, a quarter of a turn of which starts the lifting, after which the completion of the single revolution sets the injector working to boiler.

GENERAL SUGGESTIONS FOR PIPING-UP INJECTORS AND INSPIRATORS AND SUGGESTIONS THAT SHOULD BE CAREFULLY FOLLOWED WHEN MAKING PIPE CONNECTIONS.

Steam. — Connect steam pipe with highest parts of boiler and never connect with a steam pipe used for any other purpose. We would recommend a globe valve being placed in the steam pipe next to boiler which can be closed in case it is desired to take off the injector. At all other times it can be left open. When the steam connection is made, be sure and take off the injector before the steam is turned on the machine. Then blow out the steam pipe with at least forty pounds steam, which will remove all dirt and scale.

Suction. — This pipe must be tight, and if there is a valve in it the stem must be well packed.

To test the suction pipes for leaks, plug up the end of the pipe

and then screw on a common iron cap on the overflow; or if that is not at hand, unscrew cap X, and place a piece of wood on top of valve P; replace the cap and the wood will hold the valve from rising; then turn on the steam, which will locate all leaks.

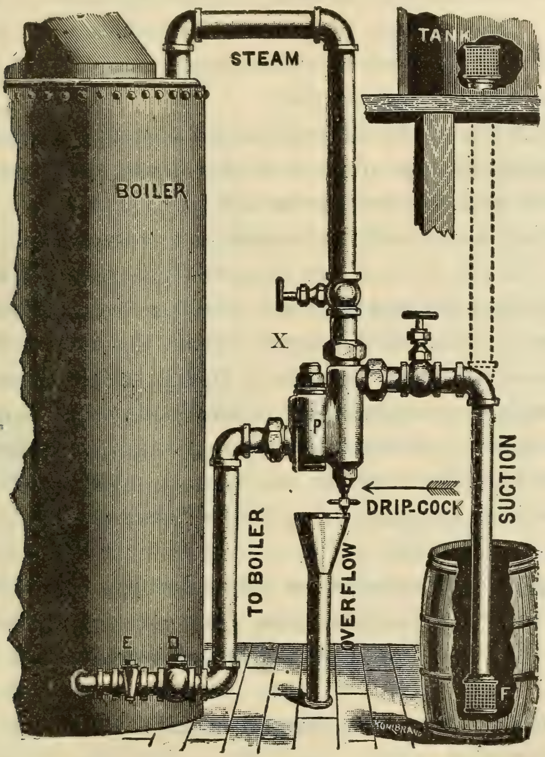


Fig. 295. Complete piping for injector.

All pipes, whether steam, suction or delivery, must be of the same or greater size than the corresponding branch of each injector. Have all piping as short and as straight as possible, and especially avoid short turns.

If any old pipe is used, see that it is not partially filled or stopped up with rust.

If the injector or inspirator has to lift the water very high or draw it very far, have the suction pipe a size or two larger than called for by the suction branch of the injector or inspirator.

Have the water supply (suction) pipe independent of any other connection. The suction pipe must be absolutely air tight; the slightest leak, in most cases, will prevent the injector or inspirator from forcing water into the boiler.

Always place a globe valve in the suction pipe as close to the injector as possible, and place it so that it will shut down against the water side and see that the stem is packed tight.

When using the injector or inspirator as NON-LIFTING, put two globe valves in the suction, one close to the injector, the other as far from it as you can conveniently, keeping the one farthest from the injector or inspirator tolerably close throttled. This will repay anyone for their trouble. The check valve may be next to boiler with a valve between it and boiler, the farther from injector the better. If the injector forces through a heater, place check valve between injector and heater. Also place a valve between heater and check valve so the latter can be taken out if necessary.

Size of pipes. — If injector or inspirator has over 10 feet lift, or a long draw, use suction pipe from strainer to valve a size larger than the connection on injector, reducing when it reaches the valve.

In all other cases, use for all pipes same size as injector connection.

Blow-off. — Always blow out steam thoroughly BEFORE CONNECTING INJECTOR, so as to remove any dirt, rust or scale that may be in the pipes.

Caution. — The suction pipe must be ABSOLUTELY TIGHT throughout. To make sure that it is so, test the suction as directed.

DIRECTIONS FOR CONNECTING AND OPERATING THE HANCOCK INSPIRATOR.

"Stationary" pattern. — Connect as shown by cut Fig. 296, showing exterior and section. For full instructions, see page 598.

For a lift of		5 ft.,	15 lbs.	steam	pressure	is	required.
"	"	10 "	20 "	"	"		"
"	"	15 "	25 "	"	"		"
"	"	20 "	35 "	"	"		"
"	"	25 "	45 "	"	"		"

Operation. — Open overflow valves Nos. 1 and 3; close forcer steam valve No. 2 and open the starting valve in the steam pipe. When the water appears at the overflow, close No. 1 valve; open No. 2 valve one-quarter turn and close No. 3 valve. The inspirator will then be in operation.

Note. — No. 2 valve should be closed with care to avoid damaging the valve seat. When the inspirator is not in operation, both overflow valves Nos. 1 and 3 should be open to allow the water to drain from it. No adjustment of either steam or water supply is necessary for varying steam pressures, but both the temperature and quantity of the delivery water can be varied by increasing or reducing the water supply. The best results will be obtained from a little experience in regulating the steam and water supply. If the suction pipe is filled with hot water, either cool off both it and the inspirator with cold water, or pump out the hot water by opening and closing the starting valve suddenly. To locate a leak in the suction pipe, plug the end, fill it with water, close No. 3 valve and turn on full steam pressure. Examine the suction pipe and the water will indicate the leak. If the inspirator does not lift the water properly, see if there is a leak in the suction pipe. Note if the steam pressure corresponds to the lift as above specified, and if the sizes of pipe used are equal in size to inspirator connections. If the inspirator will lift the water, but will not deliver it to the boiler, see if the check valve in the delivery pipe is

in working order and does not "stick." Air from a leak in the suction connections, will prevent the inspirator from delivering the water to the boiler, even more than it will in lifting it only. If No. 1 valve is damaged, or leaks, the inspirator will not work properly. No. 1 valve can be easily removed and ground.

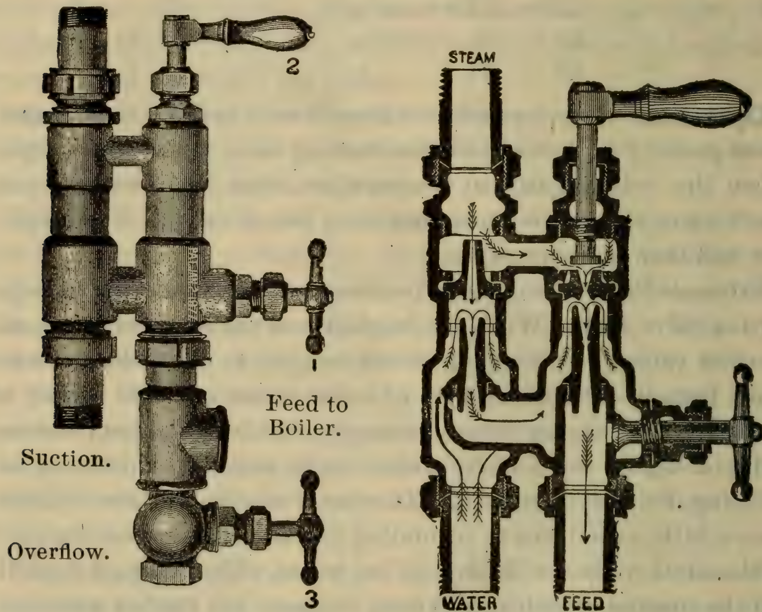


Fig. 296. Hancock inspirator.

To remove scale and deposits from inspirator jets or parts, disconnect the inspirator and plug both the suction and delivery outlets with corks. Open No. 2 valve and fill the inspirator with a solution of one part muriatic acid and ten parts water. Allow this solution to remain in the inspirator over night, then wash it thoroughly in clear water.

NOTE. — It is not generally necessary to return an inspirator for repairs. The repair parts required can be ordered and the inspirator readily put in order.

TO DISCOVER CAUSE OF DIFFICULTIES.

WHEN INJECTOR FAILS TO GET THE WATER.

1. The supply may be cut off by: (a) Absence of water at the source. (b) Strainer clogged up. (c) The suction pipe, hose or valve stopped up; or if a hose is used, its lining may be loose (a frequent cause of trouble).

2. A large leak in the suction (note that a small leak will prevent injector from working, but not from getting the water).

3. Suction pipe or water very hot. Open drip-cock, turn steam on slowly, then shut it off quickly. This will cause the cool air to rush into the suction pipe and cool it off. Repeat if necessary.

4. Lack of steam pressure for the lift; or, in some instances, too much steam pressure. If the steam pressure is very high, the injector will get the water more readily if the steam is turned on slowly and the drip-cock left open until the water is got.

IF THE INJECTOR GETS THE WATER BUT DOES NOT FORCE IT TO THE BOILER.

1. No globe valve on the suction with which to regulate the water, or else the supply water not properly regulated.

2. Dirt in delivery tube.

3. Faulty check valve.

4. Obstruction between injector and check valve, or between check valve and boiler.

5. Small leak in suction pipe admitting air to the injector along with the supply water. It is ten to one this is the cause of the difficulty every time.

6. Be sure and understand the directions for starting before condemning the injector.

THE PENBERTHY INJECTOR.

IF THE INJECTOR STARTS BUT "BREAKS."

1. Supply water not properly regulated. If too much water, the waste or overflow will be cool; if too little, the water will be very hot.

2. Leaky supply pipe admitting air to the injector. It is ten to one this is the cause of difficulty. The suction must be air tight; test as directed.

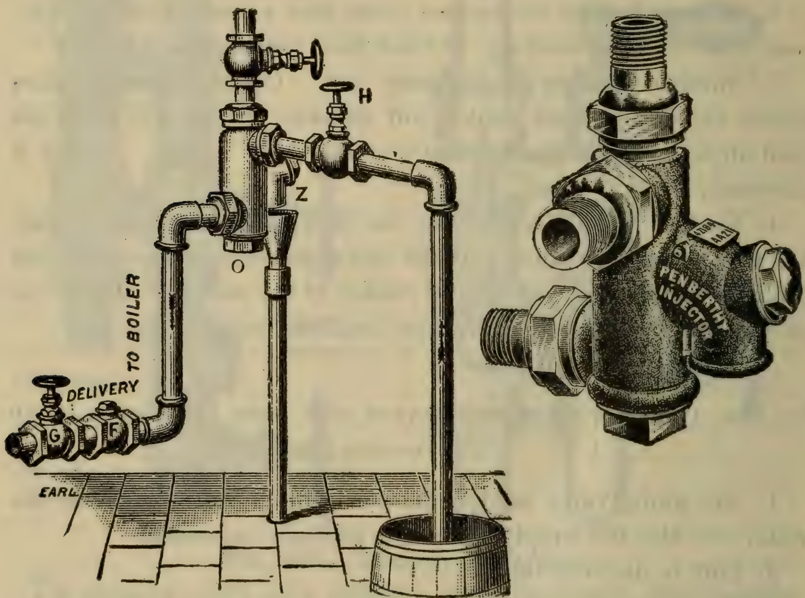


Fig. 297. Showing the Penberthy injector and pipe connections.

3. Dirt or other obstruction, such as lime, etc., in delivery tube.

4. Connecting steam pipe to pipe conducting steam to other points besides the injector, or not having suction pipe independent.

5. Sometimes a globe valve is used on the suction connection that has a loose disc, and after starting the disc is drawn down, thus partially closing the valve; it is, of course, equivalent to giving the injector too little water. To remedy this, take the globe valve off and reverse it end for end.

To clean. — To clean injector, unscrew plug O, and the removable jet inside resting in it will follow the plug out. Turn on steam (not less than forty pounds) and all dirt will be blown out. Examine all passages and drill holes and see that no dirt or scale has lodged in them. Replace jet by setting it in the plug (which acts as a guide) and screw into place tightly. Be careful not to bruise any jets, and use no wrenches on body of injector.

PRICE LIST, CAPACITY, HORSE POWER, ETC. OF PENBERTHY INJECTOR.

Size.	Price.	Pipe Connections.			Capacity per Hour. 1 to 4 ft. lift, 50 to 75 lbs. Pressure.		Horse Power.
		Steam.	Suction.	Delivery,	Maximum.	Minimum.	
OO.	$16 00	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$ in.	80 gal.	55 gal.	4 to 8
A........	18 00	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$ "	120 "	70 "	8 to 10
AA.....	20 00	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$ "	165 "	90 "	10 to 15
B	25 00	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$ "	250 "	135 "	15 to 25
BB	30 00	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$ "	340 "	165 "	25 to 35
C	40 00	1	1	1 "	475 "	300 "	35 to 50
CC.......	45 00	1	1	1 "	575 "	350 "	50 to 60
D	55 00	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$ "	750 "	400 "	60 to 95
DD	60 00	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$ "	920 "	500 "	95 to 162
E	75 00	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$ "	1300 "	700 "	120 to 150
EE.....	90 00	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$ "	1740 "	900 "	165 to 230
F.......	110 00	2	2	2 "	2270 "	1100 "	230 to 290
FF......	125 00	2	2	2 "	2820 "	1400 "	290 to 365

To test for leaks. — Plug up end of water supply pipe, then fit a piece of wood into cap Z, so that when screwed down it will hold check valve in place, then turn on steam and it will locate leak. *Do not fail to do this in case of any trouble.*

TO START AND STOP INJECTOR.

To start. — Open full the globe valve in water supply first, and then globe valve in steam pipe *wide open.* If water issues from overflow, throttle the valve H until discharge stops. Reg-

ulate injector with water supply valve, *not by steam valve.* When water supply is above the injector, in starting open *steam valve first.*

To stop. — Close the steam valve. The water valve *H* need not be closed unless the injector is used as a non-lifter, or lift is considerable.

The following table gives the number of British thermal units in a pound of water at different temperatures. They are reckoned above 32 degs. Fah., because, strictly speaking, water does not exist below 32 degs. Fah., and ice follows another law.

WATER BETWEEN 32° AND 212° FAH.

Temperature Fah.	Heat Units per pound.	Weight in lbs. per cubic foot.	Temperature Fah.	Heat Units per pound.	Weight in lbs. per cubic foot.	Temperature Fah.	Heat Units per pound.	Weight in lbs. per cubic foot.	Temperature Fah.	Heat Units per pound.	Weight in lbs. per cubic foot.
32°	0.00	62.42	110°	78.00	61.89	145°	113.26	61.28	179°	147.54	60.57
35	3.02	62.42	112	80.00	61.86	146	114.27	61.26	180	148.54	60.55
40	8.06	62.42	113	81.01	61.84	147	115.28	61.24	181	149.55	60.53
45	13.08	62.42	114	82.02	61.83	148	116.29	61.22	182	150.56	60.50
50	18.10	62.41	115	83.02	61.82	149	117.30	61.20	183	151.57	60.48
52	20.11	62.40	116	84.03	61.80	150	118.30	61.18	184	152.58	60.46
54	22.11	62.40	117	85.04	61.78	151	119.31	61.16	185	153.58	60.44
56	24.11	62.39	118	86.05	61.77	152	120.32	61.14	186	154.59	60.41
58	26.12	62.38	119	87.06	61.75	153	121.33	61.12	187	155.60	60.39
60	28.12	62.37	120	88.06	61.74	154	122.34	61.10	188	156.61	60.37
62	30.12	62.36	121	89.07	61.72	155	123.34	61.08	189	157.62	60.34
64	32.12	62.35	122	90.08	61.70	156	124.35	61.06	190	158.62	60.32
66	34.12	62.34	123	91.09	61.68	157	125.36	61.04	191	159.63	60.29
68	36.12	62.33	124	92.10	61.67	158	126.37	61.02	192	160.63	60.27
70	38.11	62.31	125	93.10	61.65	159	127.38	61.00	193	161.64	60.25
72	40.11	62.30	126	94.11	61.63	160	128.38	60.98	194	162.65	60.22
74	42.11	62.28	127	95.12	61.61	161	129.39	60.96	195	163.66	60.20
76	44.11	32.27	128	96.13	61.60	162	130.40	60.94	196	164.66	60.17
78	46.10	62.25	129	97.14	61.58	163	131.41	60.92	197	165.67	60.15
80	48.09	62.23	130	98.14	61.56	164	132.42	60.90	198	166.68	60.12
82	50.08	62.21	131	99.15	61.54	165	133.42	60.87	199	167.69	60.10
84	52.07	62.19	132	100.16	61.52	166	134.43	60.85	200	168.70	60.07
86	54.06	62.17	133	101.17	61.51	167	135.44	60.83	201	169.70	60.05
88	56.05	62.15	134	102.18	61.49	168	136.45	60.81	202	170.71	60.02
90	58.04	62.13	135	103.18	61.47	169	137.46	60.79	203	171.72	60.00
92	60.03	62.11	136	104.19	61.45	170	138.46	60.77	204	172.73	59.97
94	62.02	62.09	137	105.20	61.43	171	139.47	60.75	205	173.74	59.95
96	64.01	62.07	138	106.21	61.41	172	140.48	60.73	206	174.74	59.92
98	66.01	62.05	139	107.22	61.39	173	141.49	60.70	207	175.75	59.89
100	68.01	62.02	140	108.22	61.37	174	142.50	60.68	208	176.76	59.87
102	70.00	62.00	141	109.23	61.36	175	143.50	60.66	209	177.77	59.84
104	72.00	61.97	142	110.24	61.34	176	144.51	60.64	210	178.78	59.82
106	74.00	61.95	143	111.25	61.32	177	145.52	60.62	211	179.78	59.79
108	76.00	61.92	144	112.26	61.30	178	146.53	60.59	212	180.79	59.76

To find the number of gallons of water delivered by a steam pump in one minute, when the diameter and stroke of water piston, and the number of strokes per minute are given: —

Rule. — Square the diameter of water piston and multiply the result by .7854. Multiply this product by the stroke of the water piston in inches; and multiply this product by the number of strokes per minute, and divide the result by 231.

Example. — How many gallons of water per minute will a steam pump deliver, whose water cylinder is 6 inches in diameter and 12 inches stroke, making 60 strokes per minute?

Ans. 88.128 galls.

Operation: $6 \times 6 \times .7854 \quad 28.2744.$

And, $\dfrac{28.2744 \times 12 \times 60}{231} = 88.128.$

To find the relative proportion between the steam and water pistons.

Rule. — Multiply the area of the pump piston by the resistance of the water in pounds per square inch; and divide the product by the pressure of steam in pounds per square inch. The quotient will give the area of steam piston in square inches to balance the resistance. To this quotient add from 30 to 100 per cent of itself, — depending on the speed of the pump, — and divide the sum by .7854, and extract the square root of the quotient for the diameter of the steam piston.

Example. — What should be the diameter of the steam piston to force water against a pressure of 125 pounds per square inch, the diameter of water piston being 6 ins. and the steam pressure 60 lbs. per square inch? Ans. $10\frac{1}{2}$ inches.

Operation: $6 \times 6 \times .7854 = 28.2744$ sqr. ins.

And, $28.2744 \times 125 = 3534.3$ pounds the total resistance.

Then, $\dfrac{3534.3}{60} = 58.9$ square inches the area of steam piston.

We will add 50 per cent for friction in pump and in delivery pipe, and for a moderate speed of pump.

Then, $58.9 \times .50 = 29.45$.

And, $58.9 + 29.45 = 88.35$.

And, $\frac{88.35}{.7854} = 112.49$ sqr. ins.

Then, $\sqrt{112.49} = 10.6$ ins. the diameter of the steam piston.

To find the pressure against which a pump can deliver water, when the diameter of steam piston, pressure of steam in pounds per square inch, and diameter of water piston are given: —

Rule. — Multiply the area of steam piston by the pressure of steam in pounds per square inch, and divide the product by the area of the pump piston, and deduct from 30 to 50 per cent for friction in the delivery pipe and in the pump itself.

Example. — The area of the steam piston is 112 square inches, and the area of water piston is 28 square inches, and the steam pressure is 60 lbs. per square inch, against what pressure can the pump deliver water, the resistance from friction being 48 per cent?

Ans. 125 lbs. per sqr. in., nearly.

Operation: $\frac{112 \times 60}{28} = 240$.

And, $240 \times .48 = 115.20$.

Then, $240 - 115.20 = 124.8$.

To find the steam pressure required when the diameter of the steam piston, the diameter of the water piston, and the resistance against the pump in pounds per square inch are given: —

Rule. — Multiply the area of water piston by the resistance on the pump in pounds per square inch, and divide the product by the area of the steam piston.

Example. — The resistance against the pump, including friction, is 240 pounds per square inch. The area of steam piston is 112 square inches, and the area of water piston is 28 square inches. What pressure of steam is required to operate the pump?

Ans. 60 lbs. per sqr. in.

Operation: $\frac{240 \times 28}{112} = 60.$

Now anything over 60 lbs. will operate the pump, and the faster it is run the higher must be the pressure above 60 pounds.

To find the diameter of water piston when the diameter of steam piston, the steam pressure in pounds per square inch, and the resistance against the pump piston in pounds per square inch are given: —

Rule. — Multiply the area of steam piston in square inches by the steam pressure in pounds per square inch, and divide the product by the resistance in pounds per square inch on the water piston.

Example. — The resistance against the pump, including friction, is 240 pounds per square inch; the area of steam piston is 112 square inches, the steam pressure is 60 pounds per square inch, what should be the diameter of water piston?

Ans. 6 inches.

Operation: $\frac{112 \times 60}{240} = 35.65$ sqr. ins. Call it 36 sqr. ins.

Then, $\sqrt{36} = 6.$

To find the horse power required in a steam pump to feed a boiler with a given number of pounds of water per hour against a given pressure of steam: —

Rule. — Multiply the velocity of flow of water in feet per minute by the total pressure against which the water is pumped in pounds per square inch, and divide the product by 33,000, and the quotient will be the horse power.

Example. — What horse power is required to feed a boiler with 600 gallons of water per hour against a total resistance of 112 lbs. per square inch, including the friction in the delivery pipe, lift of water in suction pipe, weight of check valve, and friction in the pump itself? Ans. 1 H. P. nearly.

Operation: $600 \times 231 = 138,600$ cubic inches of water per hour.

And, $\frac{138,600}{60} = 2310$ cubic inches of water per minute.

And, $\frac{2310}{12} = 192.5$ feet per minute, the velocity of the water.

The total resistance is 112 lbs. per sqr. in.

Then, $192.5 \times 112 = 21560$ foot pounds.

And, $\frac{21560}{33,000} = .653$ H. P.

Now add say 50 per cent and we have $.653 \times .50 = .3265$. And, $.653 + .3265 = .9795$.

This pump will feed a boiler as shown above, or it will deliver 600 gallons of water per hour under a head of 258 feet.

Thus, $\frac{112}{.433} = 258$.

To find the horse power of boiler required to furnish steam for a pump running at its fullest capacity.

Rule. — Multiply the number of gallons of water delivered by the pump in one minute by $8\frac{1}{3}$. Multiply this product by the total height in feet to which the water is to be lifted, measuring vertically from the source of supply to the point of delivery, and divide the result by 33,000. Add from 50 to 75 per cent to the quotient for loss from friction of water in the pipe, friction in the pump, waste of steam in the cylinder, and other contingencies, and the result will give the horse power of boiler required.

Example. — What horse power of boiler is required to run a steam pump lifting 800 gallons of water per minute to a height of 163 ft. from the source of supply? Ans. 50 H. P., nearly.

Operation: $800 \times 8\frac{1}{3} = 6667$ lbs. of water.

And, $6667 \times 163 = 1,086,721$ foot pounds.

And, $\frac{1,086,721}{33,000} = 33$ H. P., nearly.

Then, $33 \times .50 = 16.50$.

And, $33 + 16.5 = 49.5$.

To find the diameter of discharge nozzle for a steam pump, when the diameter and stroke of the water piston and the number of strokes per minute are given, and the maximum flow of water in feet per minute is given: —

Rule. — Find the cubic contents of the water cylinder for one stroke in cubic feet, and multiply it by the number of strokes per minute. Multiply this product by 144 and divide the result by the velocity of the water in feet per minute, and the quotient will be the area of pump nozzle in square inches.

Example. — The diameter of water cylinder is 10 inches, and the stroke of piston is 12 inches, and the speed is 50 strokes per minute. The velocity of water required is 500 feet per minute, what should be the diameter of pump discharge nozzle?

Ans. $3\frac{1}{8}$ ins., nearly.

Operation: $10 \times 10 \times .7854 = 78.54$ sqr. ins. area of piston.

And, $78.54 \times 12 = 942.48$ cubic inches in the cylinder for one stroke.

And, $\frac{942.48}{1728} = .5454$ of a cubic foot for one stroke.

And, $.5454 \times 50 = 27.27$ cubic feet for 50 strokes per minute.

Then, $\frac{27.27 \times 144}{500} = 7.8537$ sqr. ins. the area of the nozzle.

And, $\sqrt{\frac{7.8537}{.7854}} = 3.1$ ins. the diameter.

To find the approximate size of suction pipe when its length does not exceed 25 ft. and when there are not more than two elbows in the same: —

Rule. — Square the diameter of water cylinder in inches and multiply it by the speed of the piston feet in per minute; divide this product by 200, and divide this quotient by .7854 and extract the square root, and the result will be the diameter of suction pipe, except for very small pipes when it should be made larger than the size given by the rule, in order to lessen the friction of the moving water.

Example. — The diameter of water cylinder is 6 ins., the stroke of piston is 12 ins., and the number of strokes per miuute is 60, what should be the diameter of suction pipe? Ans. 4 ins.

Operation: $\frac{6 \times 6 \times 60}{200} = 10.8.$

And, $\frac{10.8}{.7854} = 13.75.$

Then, $\sqrt{13.75} = 3.7$ ins. There is no pipe of this size made, so take 4-inch pipe.

To find the velocity in feet per minute necessary to discharge a given number of gallons of water per minute through a straight smooth iron pipe of a given diameter, regardless of friction: —

Rule. — Reduce the gallons to cubic feet and multiply by 144, and divide the product by the area of the pipe in square inches.

Example. — What should be the velocity of the water to discharge 100 gallons of water per minute through a 4-inch pipe?

Ans. 149 ft. per minute.

Operation: $\frac{100 \times 231}{1728} = 13$ cubic feet.

And, $13 \times 144 = 1872$ cubic inches placed in a continuous line.

Then, $4 \times 4 \times .7854 = 12.5664$ square inches, the area of pipe.

And, $\frac{1872}{12.5664} = 149.$

To find the velocity in feet per minute of water flowing through a pipe of given diameter, when the diameter of water cylinder and speed of piston in feet per minute are given: —

Rule. — Multiply the area of water cylinder in square inches by the piston speed in feet per minute, and divide the product by the area of the pipe in square inches.

Example. — The diameter of water cylinder is 8 ins., and the piston speed is 100 ft. per minute, and the diameter of discharge pipe is 4 ins., what is the velocity of the water in the discharge pipe? Ans. 400 ft. per minute.

Operation: $8 \times 8 \times .7854 = 50.26$ sqr. ins. area of the water piston.

And, $50.26 \times 100 = 5026$.

The area of the pipe is 12.56 sqr. ins.

Then, $\frac{5026}{12.56} = 400$.

To find the number of gallons of water discharged per minute through a circular orifice under a given head: —

Rule. — Find the velocity of discharge in feet per second and multiply it by 60, then multiply this product by the area of the orifice in square feet, and multiply this last product by 7.48, and the result will be the gallons discharged per minute.

Example. — How many gallons of water will be discharged per minute through an orifice 4 inches in diameter under a head of 81 feet? Ans. 2829.7 galls.

Operation: $\sqrt{81} = 9$. And, $9 \times 8.025 = 72.225$ feet per second, the velocity of discharge. The factor 8.025 is a constant for any head, and is found thusly: —

$\sqrt{2 \times 32.2} = 8.025$.

Or, the velocity of discharge may be found in this manner: —

$\sqrt{2 \times 32.2 \times 81} = 72.22$ feet per second, that is, the velocity in feet per second equals the square root of the acceleration

due to gravity multiplied into the head in feet. Continuing the operation, we have: —

$72.225 \times 60 = 4333.5$ feet per minute.

And, $4 \times 4 \times .7854 = 12.5664$ sqr. ins. area of orifice.

And, $\frac{12.5664}{144} = .0873$ of a square foot, the area of orifice, also.

Then, $4333.5 \times .0873 = 378.3$ cubic feet.

And, $378.3 \times 7.48 = 2829.7$ galls.

NOTE. — With a ring orifice only 64 per cent of the above amount of water would be discharged, and with a funnel-shaped orifice only 82 per cent.

To find the number of gallons of water discharged per minute under a given pressure in pounds per square inch: —

Rule. — Divide the given pressure in pounds per square inch by .433 in order to get the head in feet, and then proceed according to the foregoing rule.

Example. — How many gallons of water will be discharged per minute through an orifice one square inch in area, under a pressure of 35.073 lbs. per square inch? Ans. 81 galls. per minute.

Operation: $\frac{35.073}{.433} = 81$ ft., head equivalent to the given pressure.

And, $\sqrt{2 \times 32.2 \times 81} = 72.225$ ft. per second the velocity.

And, $72.225 \times 60 = 4333.5$.

Also, $\frac{1}{144} = .00694$ of a square foot, equals the area of the orifice.

And, $4332.5 \times .00694 = 30.07449$.

And, $30.07449 \times 7.48 = 224.9$ galls.

Then, deducting 64 per cent, we have: —

$224.9 \times .64 = 143.9$.

And, $224.9 - 143.9 = 81$.

To find the area of orifice in square ins. necessary to discharge a given number of gallons of water per minute under a given head in feet: —

Rule. — Divide the number of gallons by the constant number 15.729 multiplied into the square root of the head, and the result will be the area of orifice in square inches.

Example. — What must be the area of orifice to discharge 1778.5 gallons of water per minute under a head of 81 feet?

Ans. 12.56 sqr. ins.

Operation: $\sqrt{81} = 9$.

And, $9 \times 15.729 = 141.6$.

Then, $\frac{1778.5}{141.6} = 12.56$.

To find how many gallons of water will flow through a straight smooth iron pipe in one minute under a given pressure in pounds per square inch, or head in feet: —

Rule. — Multiply the inside diameter of the pipe in feet by the head in feet, and divide the product by the length of pipe in feet. Extract the square root of the quotient and multiply it by 48, and the product will be the velocity of flow in feet per second. Multiply this result by 12 to reduce it to inches, and by 60 for the flow per minute, and multiply again by the area of the pipe in square inches, and divide by 231 for the gallons discharged per minute.

Example. — How many gallons of water will be discharged per minute through a 4-inch pipe 2000 feet long, under a head of 92 feet?

Ans. 230 galls. per minute.

Operation: 4 ins. = .33 of a foot.

And, $92 \times .33 = 30.36$.

And, $\frac{30.36}{2000} = .015$.

And, $\sqrt{.015} = .1225$.

Then, $.1225 \times 48 \times 12 = 70.56$ ins. per second.

And, $70.56 \times 60 = 4233.60$ ins. per minute.

Then, $4 \times 4 \times .7854 = 12.56$ sqr. ins. the area of the pipe.

And, $4233.60 \times 12.56 = 53174.016$ cubic ins.

Then, $\frac{53174.016}{231.} = 230.2$.

Example. — Assume two wells A and B with their mouths on a level. Well A is 26 ft. deep, and well B is 40 ft. deep. Well A is fed by natural springs and has a depth of water of 5 feet. The distance between the wells is 600 feet. How many gallons of water will a 1 inch pipe, laid perfectly straight and level, syphon over in one minute providing well B is always pumped dry, and that the pipe extends into well A 26 feet, and into well B 38 feet, using bends instead of elbows?

Ans. 4 galls. per minute.

Operation. — The head equals 38 feet.

The diameter of the pipe equals .0833 foot.

Then, $600 + 38 + 26 = 664$ ft. total length of pipe.

And, $38 \times .0833 = 3.1654$.

And, $\frac{3.1654}{664} = .0047$.

And, $\sqrt{.0047} = .068$.

Then, $.068 \times 48 = 3.264$ ft. velocity per second.

And, $3.264 \times 60 = 195.840$ ft. velocity per min.

The area of pipe equals .7854 sqr. inch.

Then, $195.840 \times .7854 = 153.8127$.

And, $153.8127 \times 7.48 = 1150.52$.

And, $\frac{1150.52}{144} = 8$ nearly, gallons.

Deducting 50 per cent on account of 2 bends and friction, we have 4 gallons per minute syphoned over.

To find the head in feet due to friction in a pipe running full: —

Rule. — Multiply the length of the pipe in feet by the square of the number of gallons per minute, and divide the product by 1,000 times the 5th power of the diameter of the pipe in inches. The quotient less 10 per cent is the head in feet necessary to overcome the friction.

NOTE. — The head is the vertical distance from the surface of the water in the tank or reservoir, to the center of gravity of the lower end of the pipe, when the discharge is into the air, or, to the level surface of the lower reservoir when the discharge is under the water.

Example. — A 2-inch pipe 100 feet long and running full, discharges 50 gallons of water per minute, what is the head in feet due to friction? Ans. 7.029 feet.

Operation: $2 \times 2 \times 2 \times 2 \times 2 = 32 =$ the 5th power of the diameter of the pipe.

And, $50 \times 50 = 2500$.

And, $2500 \times 100 = 250,000$.

Also, $32 \times 1,000 = 32,000$.

Then, $\frac{250,000}{32,000} = 7.81$.

And, 7.81 less 10 per cent of itself equals 7.029.

The resistance to the flow of water in pounds per square inch, due to friction, is found by dividing the friction head by 2.3.

Thus, $\frac{7.029}{2.03} = 3.05$ lbs.

To find the size of pump required to feed a boiler of a given capacity: —

Rule.—Multiply the number of pounds of water evaporated per pound of coal by the number of pounds of coal burned per sqr. foot of grate surface per hour, and multiply this product by the number of square feet of grate surface in the boiler furnace. This will give the number of pounds of water evaporated by the boiler in one hour. Divide this by 60 to find the evaporation per minute, and divide again by $8\frac{1}{3}$ in order to get the evaporation in gallons per minute; add from 10 to 15 per cent to the last result for leakage and other contingencies, and select a pump that will deliver the gross number of gallons of water per minute at any speed that may be desired, usually taken, however, at fifty feet per minute.

Example. — What should be the dimensions of the water end of a steam pump, and what should be the speed of piston to supply a boiler having a grate surface of 20 square feet, and burning 15 pounds of coal per square foot of grate, and evaporating 9 pounds of water per pound of coal per hour?

Operation: $20 \times 15 \times 9 = 2700$ pounds of water evaporated per hour.

And, $\frac{2700}{60} = 45$ lbs. of water evaporated per minute.

And, $\frac{45}{8\frac{1}{3}} = 5.4$ galls. per minute.

Then, 5.4 plus 10 per cent of itself, equals 6 galls. nearly per per minute.

Referring to a pump maker's catalogue we find that a single pump $3\frac{1}{2}'' \times 2\frac{1}{4}'' \times 5''$, making 90 strokes per minute, will do the work, or, a duplex pump $3'' \times 2'' \times 3''$, making 100 strokes per minute will do the work equally as well. Again, adding 10 per cent to the pounds of water evaporated per minute we have, $45 + 4.5 = 49.5$ pounds. And, $49.5 \times 27.71 = 1371.64$ cubic inches displacement in the water cylinder per minute, and at 90 strokes per minute we have 15.24 cubic inches displacement per stroke.

Thus, $\frac{1371.64}{90} = 15.24$, which is all that is required for our boiler.

Now, taking the above single pump we have: $2.25 \times 2.25 \times .7854 \times 5 = 19.8$ cubic inches displacement per stroke. And, taking the duplex pump we have: $2 \times 2 \times .7854 \times 3 \times 2 = 18.8$ cubic ins. displacement for each double stroke of the piston, or, plunger, showing that either pump is of ample capacity to feed the boiler at a fair piston speed.

To find the duty of a pumping engine when the number of pounds of coal burned, the number of gallons of water pumped, the pressure in pounds per square inch against which the pump piston works, and the height of suction are given: —

Rule. — Find the head in feet against which the pump works, by multiplying the pressure by 2.3, add the suction in feet to this head in order to get the total head. Multiply the gallons of water by $8\frac{1}{3}$ to get the pounds of water delivered. Then multiply the total number of pounds of water by the head in feet, and divide the product by the number of pounds of coal divided by 100, and the result will give the duty in foot pounds. The duty of a pumping engine is the number of pounds of water raised one foot high for each 100 pounds of coal burned.

Example. — What is the duty of an engine pumping 2,890,000 gallons of water in 12 hours against a pressure of 30 pounds per sqr. inch, the suction being 12 feet, and coal burned 24,470 pounds? Ans. 8,070,426 foot pounds.

Operation: $30 \times 2.3 = 70$ nearly the head in feet.

And, $2{,}890{,}000 \times 8\frac{1}{3} = 24{,}083{,}333$ pounds of water.

Also, $70 + 12 = 82$ ft. total lift of water.

And, $24{,}083{,}333 \times 82 = 1{,}974{,}833{,}306$ lbs. of water lifted one foot high in 12 hours.

Then, $\frac{24{,}470}{100} = 244.7$.

And, $\frac{1,974,833,306}{244.7} = 8,070,426.$

To find the horse power of a pumping engine: —

Rule. — Divide the number of pounds of water raised one foot high in one minute by 33,000.

Example. — What is the H. P. of the pumping engine given in the above example? Ans. 83.11 H. P.

Operation: $12 \times 60 = 720$ minutes.

And, $\frac{1,974,833,306}{720} = 2,742,824$ lbs. of water raised one foot high in one minute.

Then, $\frac{2,742,824}{33,000} = 83.11.$

To find the capacity of a pump to feed a boiler it is necessary to know how much water the boiler is capable of evaporating per minute or per hour. Each horse power of boiler capacity corresponds to an evaporation of thirty pounds of water per hour. It is good practice to operate a pump slowly and continuously, and for this reason the pump running at its normal speed should be capable of supplying about twice as much water as the boiler evaporates under usual conditions.

To find the diameter of water cylinder to deliver a certain number of gallons of water per minute, when the stroke of the piston and the number of strokes per minute are given: —

Rule. — Multiply the number of gallons by 231, and divide the product by the stroke of the piston, and divide this quotient by the number of strokes per minute, and divide this last quotient by .7854, then extract the square root of the result for the diameter of the water piston.

Example. — A battery of boilers evaporate 100,000 pounds of water in one hour, what should be the diameter of water cylinder to supply this battery, the stroke of piston being 12 inches and making 100 strokes per minute? Ans. 7 inches.

Operation: $\frac{100,000}{60} = 1666\frac{2}{3}$ pounds of water evaporated in one minute.

And, $\frac{1666\frac{2}{3}}{8\frac{1}{3}} = 200$ galls. evaporated in one minute. Then following the above rule we have: —

$200 \times 231 \quad = 46200.$

And, $\frac{46200}{12} = 3850.$

And, $\frac{3850}{100} = 38.5.$

And, $\frac{38.5}{.7854} = 49.$

Then, $\sqrt{49} = 7''$ the required diameter.

To determine the H. P. of boiler a steam pump of given dimensions will supply when the number of strokes per minute are given: —

Rule. — Multiply the area of the piston is square inches by the stroke of piston in inches, and this product divided by 231 will give the gallons per stroke. Multiply this quotient by the number of strokes per minute for the number of gallons per minute, and by 60 for the number of gallons per hour. Multiply this product by $8\frac{1}{3}$ to find the number of pounds of water per hour delivered by the pump, and divide this product by 30 for the H. P. of boiler the pump will supply. This rule is based upon the assumption that the full capacity of the water cylinder is delivered at each stroke, no allowance being made for slippage, leakage, or short strokes.

Example. — The water piston of a steam pump is 6 inches in diameter and has a stroke of 12 inches, making 100 strokes per minute, what H. P. of boiler will the pump supply?

Ans. 2448 H. P.

Operation: $6 \times 6 \times .7854 = 28.2744$ sqr. ins. area of piston.

And, $28.2744 \times 12 = 339.2928$ cubic inches for one stroke.

And, $\frac{339.2928}{231} = 1.4688$ galls. per stroke.

And, $1.4688 \times 100 = 146.88$ galls. per minute.

And, $146.88 \times 60 = 8812.8$ galls. per hour.

And, $8812.8 \times 8\frac{1}{3} = 73{,}440$ pounds of water per hour.

Then, $\frac{73440}{30} = 2448$ H. P. of boilers.

Watt allowed one cubic foot ($62\frac{1}{2}$ lbs.) of water per H. P. per hour. Then taking this allowance instead of 30 as above, we would have, $\frac{73440}{62.5} = 1175$ H. P. of boilers which the above pump would be suitable for, and which could be run very slowly, thus prolonging the life of the pump.

Even though a suction pipe should be perfectly air tight, a perfect vacuum cannot be formed in it, because water contains air, and even the coldest water gives off some vapor tending to impair the vacuum. Twenty-eight feet is a very good lift for a pump taking its water by suction.

Pump Formulas.

Gals. per Min. $= .0034 \times$ Diameter$^2 \times$ Stroke in ins. $\times$ No. of Strokes.

Sq. of Diam. $=$ Gals. per Min. $\div$ ($.0034 \times$ Stroke in ins. $\times$ No. of Strokes).

Length of Stroke $=$ Gals. per Min. $\div \frac{\text{Square of Diameter} \times 34}{\text{No. of Strokes}}$

No. of Strokes $=$ Gals. per Min. $\div \frac{\text{Square of Diameter} \times 34}{\text{Length of Stroke}}$

CHAPTER XXI.

MECHANICAL REFRIGERATION.

About the first thing asked by persons who are becoming interested in the subject of refrigerating and ice-making is, "Tell me how the thing is done?"

Mechanical refrigeration, primarily, is produced by the evaporation of a volatile liquid which will boil at low temperature, and by means of a special apparatus the temperature and desired amount of refrigeration is placed under control of the operator.

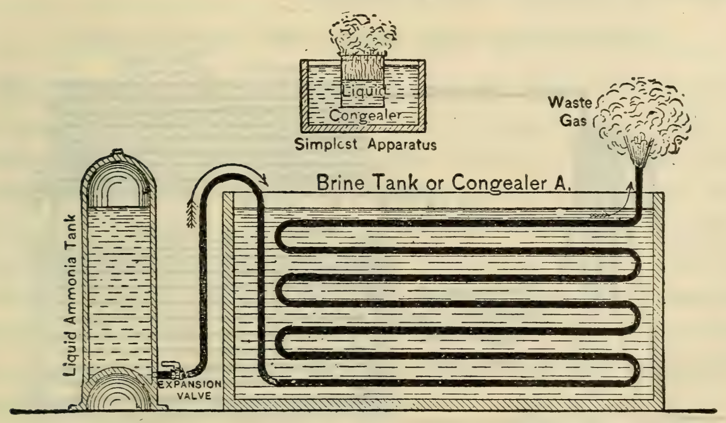


Fig. 298. Elememtal refrigerating apparatus.

The simplest form of refrigerating mechanical apparatus consists of three principal parts: *A*, an "evaporator," or, as sometimes called, a "congealer," in which the volatile liquid is vaporized; *B*, a combined suction and compressor pump, which

sucks, or properly speaking, "aspirates" the gas discharged by the compressor pumps, and under the combined action of the pump pressure and cold condenser, the vapor is here reconverted into a liquid, to be again used with congealer. We now see the function of the compressor pumps and condensers.

PRINCIPLES OF OPERATION.

The action of all refrigerating machines depends upon well-defined natural laws that govern in all cases, no matter what type of apparatus or machine is used, the principle being the same in all; while processes may slightly vary, the properties of the particular agent and manner of its use affecting, of course, the efficiency or economic results obtained.

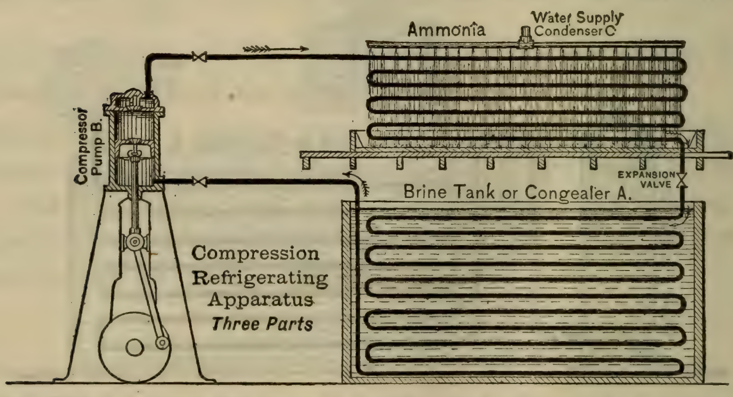


Fig. 299. Outline drawing of mechanical compression system.

OPERATION OF APPARATUS.

See Fig. 299. The apparatus being charged with a sufficient quantity of pure ammonia liquid, which we will, for simplicity, assume to be stored in the lower part of the condenser C, a small cock or expansion valve controlling a pipe leading to the congealer

or brine tank A, is slightly opened, thus allowing the liquid to pass in the same office as a tube or flue in steam boiler and having precisely the same function, it may be called heating or steam making service. The amount of water capable of being boiled into steam in a boiler depends upon the square feet of heating surface, temperature of fire and pressure of steam; and the same is true of the capacity of heating surface presented by the coils in the evaporator. The heat is transmitted through the coils from surrounding substance to the ammonia liquid, which is boiled into a vapor the same as water is boiled into steam in a steam boiler;. as previously explained, the heat thus becomes cooler; the amount taken up and made negative being in proportion to the pounds of liquid ammonia evaporated.

FUNCTION OF THE PUMP AND CONDENSER.

The office of the compressor, pump and condenser is to reconvert the gas after evaporation into a liquid, and make the original charge of ammonia available for use in the same apparatus, over and over again. It will appear to the reader, after having carefully followed the text, that the pump and condenser might be dispensed with, but these conditions may only be economically realized when the, at present, expensive ammonia liquid can be obtained in great quantities and at less cost than the process of reconverting the vapor into a liquid by compression machinery and condenser on the spot.

WHAT DOES THE WORK.

The real index of the amount of cooling work possible is the number of pounds of ammonia evaporated between the observed range of temperature. To make the above clear, we will add that each pound of ammonia during evaporation is capable of storing up a certain quantity of heat, and that the simplest forms

of refrigerating apparatus might consist, as shown by engraving, of two parts, to wit: A congealer and a tank of ammonia. In this apparatus the ammonia is allowed to escape from the tank into the congealer as fast as the coils therein are capable of evaporating the liquid into a gas. When completely evaporated the resulting vapor is allowed to escape into the atmosphere, which means it is wasted, the supply being maintained by furnishing fresh tanks of ammonia as fast as contents are exhausted. This process, while simple, would be tremendously expensive, costing at the rate of about $200 per ton, refrigerating or ice-melting capacity. To recover this gas and reconvert to a liquid on the spot in a comparatively inexpensive manner, is the object to be obtained.

MECHANICAL COLD EASILY REGULATED.

This being under the control of the cock or valve leading from the condenser called an expansion valve. As the gas begins to form in the evaporator, the compressor pump B is set in motion at such a speed as to carry away the gas as fast as formed, which is discharged into the condenser under such pressure as will bring about a condensation and restore the gas to the liquid state; the operation being continuous so long as the machinery is kept in motion.

UTILIZING THE COLD.

To utilize the cold thus produced for refrigerating, two methods are in use, the first of which is called the brine system; the second is known to the trade as the direct expansion system, both of which systems will be explained at some length.

BRINE SYSTEM.

In this method, the ammonia evaporating coils are placed in a tank, which is filled with strong brine made of salt, which is well known not to freeze at temperature as low as zero. This is the brine

tank or congealer *A*. The evaporating or expansion of the ammonia in these coils robs the brine of heat, as heretofore explained, the process of storing cold in the brine going on continuously and being regulated, as required, at the gas expansion valve. To practically apply the cold thus manufactured, the chilled brine or non-freezing liquid is circulated by means of a pump through coils of pipe which are placed on the ceilings or sides of the apartments to be refrigerated, the process being analogous to heating rooms by steam.

THE BRINE COOLS THE ROOMS.

The cold brine in its circuit along the pipes becomes warmer by reason of taking up the heat of the rooms, and is finally returned to the brine tank, where it is again cooled by the ammonia coils, the operation, of course, being a continuous one.

DIRECT EXPANSION SYSTEM.

By this method, the expansion or evaporating coils are not put in brine tanks, but are placed in the room to be refrigerated, and the ammonia is evaporated in the coils by coming in direct contact with the air in the room to be refrigerated, no evaporating tank being used.

RATING OF THE MACHINE IN TONS CAPACITY.

For the information of the unskilled reader, we will state that machines are susceptible of two ratings; that is, either their capacity is given in tons of ice they will produce in one day (24 hours), called ice-making capacity; or they are rated equal to the cooling work done by one ton of ice-making per day (24 hours), called refrigerating capacity.

DIFFERENCE IN THESE RATINGS.

Ordinarily the ice-making capacity is taken at about one-half of the refrigerating capacity, but this is only approximate, and

INSTRUCTIONS FOR OPERATING REFRIGERATING AND ICE-MAKING MACHINERY.

First, a competent engineer should be placed in charge of the plant, and he should be held responsible for the performance of the plant.

He should have charge of all help in the engine room, tank-room, and all men who work around the plant.

He should acquaint himself with all pipe and valves about the plant, so that in case of trouble. he will know what to do.

Valves should be provided in suitable places, so that if it becomes necessary, he can transfer the ammonia from one part of the plant to another; before attempting to transfer the ammonia, the engineer in charge should carefully see that he fully understands where the ammonia is to be put, and that there is sufficient space to contain same. In making transfers, always run the machine very carefully.

When starting the engine, start slowly, and before the pumps show a vacuum in the coils, open the regulating valve slightly, and then speed up the engine gradually.

The back pressure should be kept at about 15 pounds above zero, depending on the frost shown on suction pipe. When the machine is run to its full capacity, it should show frost on the suction pipe just above the brine tank; in some machines, it is necessary to freeze back to machine, as the frost takes up the heat of compression and takes the place of the water jacket.

In general, the engineer in starting the machine, will first see that the discharge valves are open, then start the machine slowly, and open the suction valves slowly. When the machine is fully up to speed, watch the gauges and note the pressure. The condenser pressure should be somewhere between 150 and 180 lbs.

depending upon the temperature of the water and atmosphere. The suction or back pressure should be about 20 pounds, unless the temperature of the brine in the tanks is below 18 degrees ; if it is, the low pressure should be reduced in proportion ; the frost on the suction pipe will determine the back pressure to be carried. It is advisable to use as high a back pressure as possible without frosting back.

Regulate the low pressure by means of the feed valves. The pressure should rise slowly ; watch carefully to see that the feed valves work regularly so that each valve may supply the proper amount of ammonia to the coils.

The frosting of the valves will indicate how they are working, and a little practice will enable the engineer to judge of this.

When the ammonia disappears from the liquid receiver, it is possible that too much feed has been given the coils, and that they are flooded ; it is well to shut off the main liquid valve to the tanks and pump them out until the back pressure falls to about five pounds ; the valve can then be opened and the regular operation resumed.

This pumping down is for the purpose of getting the ammonia all out of the tank coils, where it will sometimes lie ; when this is the case, the suction line to the machine will be frosted, even when the tanks are not below temperature of 18 degrees ; when the tanks are below 18 degrees, frost will generally appear in the suction line to the machine.

In case of a leak in the pipe, the connections are made to the pumps so that the large valves on the suction and discharge pipes may be closed, and the small by-pass valves opened and by running the machine slowly, the discharge pipe system can be pumped into the suction and brine tanks. In case of a leak in the suction side, it is only necessary to shut off the liquid valve and pump all of the ammonia up into the condenser and keep it there by shutting the valve.

When shutting down a machine, close the liquid valve on the

brine tank, and run the machine until the pressure is brought down to zero on the gauge. Do not create a vacuum, because air is liable to leak into the pumps through the stuffing boxes.

It is well to watch the compressor carefully for leaks; a leak into the cylinder either through leaky discharge valves, the gaskets or the cylinder head gasket between the cylinder and the discharge port, will materially reduce the compressor capacity. It will not take long for a compressor to waste a ton of coal.

If the engineer in charge of a compressor has a chance, he should take off the cylinder head and examine the cylinder gasket. If it looks bad, replace it with a new one; rub the valves to their seats with flour of emery and oil, and see that they have a good bearing; also examine the valve cage gaskets.

A good test for a compressor, to see whether all connections are tight about the compressor, is to connect a pressure gauge to the indicator connection; compress the gas in the cylinder so as to have a high pressure. Note the pressure on the gauge. If it does not decrease, the compressor and connections are tight. If it decreases rapidly, either the valves need regrinding, or the piston needs new rings, or the cylinder should be rebored, or all these troubles may exist at the same time.

All pipe and fittings between the machine and condenser, should be looked after as all flanges are provided with lead gaskets, and these flanges should be examined occasionally, and when the plant is shut down and allowed to cool, the flanges should be tightened up.

The pipes of the condensers should be kept as clean as possible, so that the water will flow evenly over all pipes alike, in order to extract as much heat as possible from the ammonia.

The colder the ammonia can be kept in the condenser, the

more work it will do in the tanks; on this account, well water is to be preferred.

The oil trap on the line from the machine to the condenser, should be examined once every ten days and the oil drawn off.

If the system gets clogged with oil, and where there is more than one tank, pump the brine into the other tanks and when the brine is all out of the tank, disconnect the coil at top and bottom, connect steam, attach a gauge, and then drive a pine plug in bottom of coil and put about 30 pounds of steam on coil for ten or twenty minutes; on the bottom of the return header is a purge cock that must be opened frequently, while the steam is on coil. Then knock plug out of coil at bottom, and let the steam come through coil for a short time, and then disconnect the steam and connect the air pump to the coil and put on thirty pounds air pressure for ten minutes; all coils must be treated in the same way.

If there is only one tank, the coils must be taken out of tank and blown out, or the brine pumped into the cans. At the same time, the expansion valves must be overhauled. After all coils and headers are connected the whole should be tested for leaks, first, under air pressure, and second, with ammonia and a sulphur stick while the tank is empty.

The sulphur sticks are prepared by dipping a stick of wood into melted sulphur. The sticks are then burned close to where leaks are suspected. Any leak is at once indicated by a white smoke.

STEAM CONDENSERS.

Steam and ammonia condensers should be kept clean and free from all scale, and should be supplied with sufficient water to condense the steam and ammonia.

THE REBOILERS.

The reboilers are for the purpose of removing any air or gases which may be in the steam. The water in the reboilers should at all times be kept boiling, for if it is not, the ice will be white. The regulating valves on the reboilers should be examined often, to see that they are not sticking. A steam connection should be provided in the flat coolers, so that after shutting off the water, which flows over the cooler, and the water from the reboilers, steam may be blown through it from the top downward and out at the bottom of the coil; this should be done frequently, land it will be found to save the filters.

The oil separator should be examined at least once a year and cleaned. The object of this separator is to remove the oil from the steam, and, if working properly, there should be a small stream of water and oil flowing from the drain.

AIR IN THE SYSTEM.

When air or permanent gases are in the system, it will be manifested by unusually high condenser pressure, and the efficiency of the machine will be reduced. To remove the air, attach a bent piece of one-fourth inch pipe to the small valve placed on the top of the condenser; place the other end of the pipe in a bucket of water; open the valve slightly, and if air is present, it will bubble up through the water, while the ammonia will produce a crackling noise, the same as when steam is turned into water.

GASES IN THE PLANT.

Accumulation of gases in a plant sometimes consists of atmospheric air, but sometimes also of hydrogen and nitrogen, due to the decomposition of ammonia.

The best way to remove these gases from the system, is by drawing them off at the top of the ammonia condenser coils, where a small valve will be found on top of each coil where a small pipe may be attached to the small valves ; put the other end in a bucket of water. If, on opening the valve, bubbles are seen to escape through the water, the valve should be kept open as long as such bubbles appear ; when a crackling noise is heard in the water, close the valve.

Every engineer should keep himself posted as to the exact clearance existing between the piston and the head or heads of the compressor. A convenient way to ascertain this, is to remove one of the valves from the compressor head and insert a piece of soft lead rolled in the shape of a wire through the valve chamber into the cylinder and pinch the machine over until the piston of the compressor squeezes the lead against the head. When the lead is withdrawn, the exact clearance will be shown.

If the valves pound, it may be that the valves have too much lift. Try a stiffer spring. A spring that is too stiff will also cause valves to pound.

Never create a vacuum in an ammonia system, unless it is absolutely necessary to repair a leak or for similar purposes ; it will have a tendency to admit air into the system ; the air must be kept out ; pump to zero on the gauge, but no lower.

Never put any ammonia in a plant until it has been tested ; this can be done by drawing a sample out of the drum and seeing that it will all evaporate and leave no residue.

Keep a record of the temperatures of brine and condensing water, and evaporating and condensing pressures.

It is a good plan to have a duplicate set of valves on hand, all ready to replace any leaky valve.

A leaky suction valve is sometimes the cause of considerable loss in capacity. It can be located by heat on the suction connection or by the hiss that is ever present when a valve leaks, or by the appearance of the valve.

Test the oil that is being used in the compressors by subjecting a sample to low temperature — get a bottle of the oil and cover with fine ice and salt; the result will demonstrate whether it will stand a low temperature or not. If the oil gets thick and gummy, and a separation occurs, leaving a thin transparent, watery liquid, in which the heavy part of the oil settles, and which gives off an odor like benzine. Reject the oil, as it will produce gases in the system, and give trouble.

Oil may be tested with ammonia gas; animal fat will saponify when subjected to alkali tests.

In ice plants, the engineer should see that the ice is always pulled regularly. The distilled water is supplied regularly and it should be used in the same way; pull but one can and refill before pulling the next.

Keep the brine in the tanks over the top pipe; at the end of the season, when the plant is shut down, leave the cans in the brine for if the cans are taken out, the brine will be lowered and where the pipes are exposed, they will rust very fast; keep the tank covers clean. The strength of the brine should be about 80 on a saltometer.

If ice forms on the coils, the brine is weak, and the brine must be strong enough so it will not freeze.

In making repairs to coils, while immersed in brine the workmen should besmear their arms and hands with cylinder oil, or lard or tallow, as that will enable them to keep them in the brine longer.

In case the temperature of the brine rises above 30 degrees, do not attempt to reduce the temperature without first examining the cans to see whether the ice has thawed loose from them; in case it has thawed, pull all ice and refill the cans before reducing temperature; if this is not done, the freezing of the water around the ice in the cans will burst them.

TESTING FOR WATER BY EVAPORATION.

As shown by the engraving, screw into the ammonia flask a piece of bent one-quarter inch pipe, which will allow a small bottle to be placed so as to receive the discharge from it. This test bottle should be of thin glass with wide neck, so that quarter-inch pipe can pass readily into it, and of about 12 cubic inches capacity. Put the wrench on the valve and tap it gently with a hammer. Fill the bottle about one-third full and throw sample out in order to purge valve, pipe and bottle. Quickly wipe off moisture that has accumulated on the pipe, replace the bottle and open

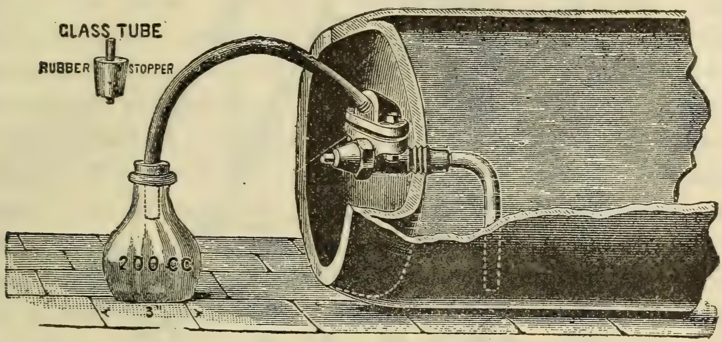


Fig. 300. Showing connections to flask.

valve gently, filling the bottle about half full. This last operation should not occupy more than one minute. Remove the bottle at once and insert in its neck a stopper with a vent hole for the escape of the gas. A rubber stopper with a glass tube in it is the best, but a rough wooden stopper, loosely put in, will answer the purpose. Procure a piece of solid iron that should weigh not less than eight or ten pounds, pour a little water on this and place the bottle on the wet place. The ammonia will at once begin to boil, and in warm weather will soon evaporate. If any residuum, pour it out gently, counting the drops carefully. Eighteen drops are about equal to one-tenth of an ounce, and if the sample taken

amounts to, say, 6 cubic inches, we can readily approximate the percentage of the liquid remaining.

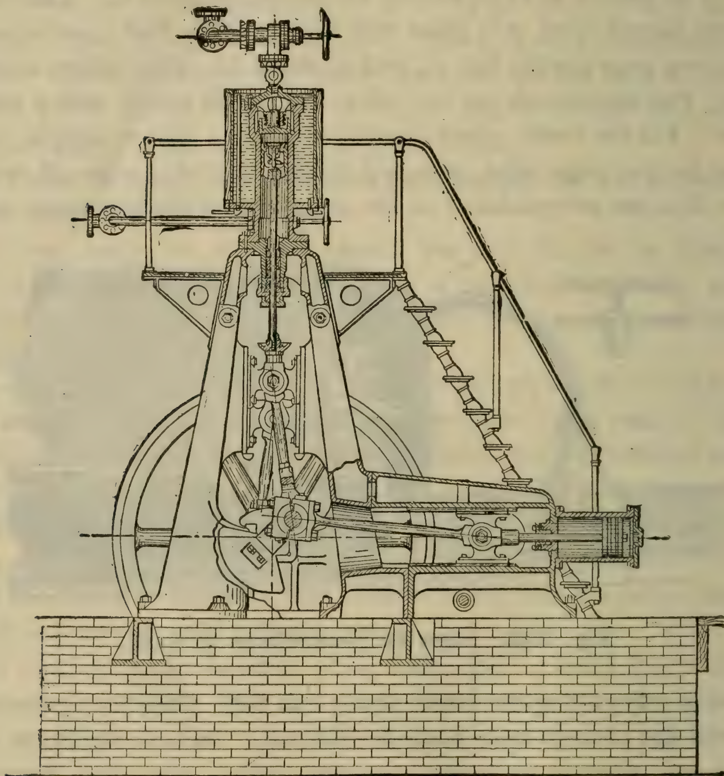


Fig. 301. Sectional view of 10-ton refrigerating machine.

LUBRICATION OF REFRIGERATING MACHINERY.

It is well to speak of this, for the reason that it is an important subject; and some users of machinery think that a cheap, low

grade of oil is really the cheapest. To disabuse their minds of this idea and suggest the necessity of high grade oils, both on the score of economy and to keep the machinery at all times in efficient running order, we suggest the following: First-class refrigerating machinery calls for the use of at least three different kinds of oil, Nos. 1, 2 and 3, each of high grade: —

No. 1. For use in the steam cylinder, and is known in the trade as cylinder oil. This ranges in price from 50c. to $1 per gallon. Good cylinder oil should be free from grit, not gum up the valves and cylinder, should not evaporate quickly on being subjected to heat of the steam, and when cylinder head is removed, a good test is to notice the appearance of the wearing surfaces; they should be well coated with lubricant which, upon application of clean waste, will not show a gummy deposit or blacken. Use this oil in a sight feed lubricator with regular feed, drop by drop.

No. 2. For use of all bearing and wearing surfaces of machine proper — an oil that will not gum, not too limpid, with good body, free from grit or acid and of good wearing quality, flowing freely from the oil cups at a fine adjustment without clogging, and a heavier grade should be used for lubricating the larger bearings.

No. 3. For use in compressor pumps. This oil should be what is called a cold test, or zero oil, of best quality.

Best paraffine oil is sometimes used; as also a clear West Virginia crude oil. This oil, when subjected to a low temperature, should not freeze.

EFFECTS OF AMMONIA ON PIPES.

Ammonia has no chemical effect upon iron; a tank, pipe or stop-cock may be in constant contact with ammonia for an indefinite time and no action will be apparent. The only protection, therefore, that ammonia-expanding pipes require is from corrosion on the outer surface. As long as the pipes are covered

with snow or ice, corrosion does not occur; the coating of ice thoroughly protects them from the oxidizing effect of the atmosphere; but alternate freezing and thawing requires protected surfaces, which are best obtained by applying a coat of paint every season.

Expansion coils having to withstand but a maximum working pressure of thirty pounds per square inch, are constructed with such absolute security, in whole and in detail, as to make them one of the most perfect pipe constructions on a large scale ever applied in practice.

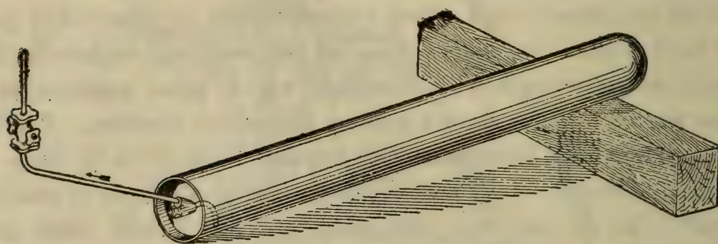


Fig. 302. Position of tank to be emptied.

TO CHARGE THE SYSTEM WITH AMMONIA.

Position of the tank should be as shown, the outlet valve pointing upwards and the other end of the tank raised 12″ to 15″. The connection between the outlet valve of the tank and the inlet cock of the system should be a $\frac{3}{8}''$ pipe. In charging, open valve of the tank cautiously to test connection; if this is tight, open valve fully; start machine and run slowly till tank is empty. The tank is nearly empty when frost begins to appear on it; run the machine till suction gauge reaches atmospheric pressure. If it holds at this pressure when machine is stopped, the tank is empty; if not, start up again. In disconnecting, close the valve on the tank first, the inlet cock of the system. Weigh tank

before and after emptying; each standard tank contains from 100 to 110 pounds of ammonia.

PROCESS OF MECHANICAL REFRIGERATION.

The process of mechanical refrigeration is simply that of removing heat, and mechanism is necessary, because the rooms and articles from which the heat is to be removed are already as cold, or colder than their surroundings, and consequently, the natural tendency is for the heat to flow into them instead of out of them. The fact that a body is already *cold* does not prevent the removal of more heat from it and making it still colder. The term cold describes a sensation and not a physical property of matter; the coldest bodies we commonly meet with are still possessed of a large quantity of heat, part of which, at least, can be abstracted by suitable means. The only means by which heat can be removed from a body is to bring in contact with it a body colder than itself. This is the function that ammonia performs in mechanical refrigeration. It is so manipulated as to become colder than the body we wish to cool. The heat thus abstracted by it is got rid of by such further manipulation that (while still retaining the heat it has absorbed) it will be hotter than ordinary cold water, and therefore, part with its heat to it. Ammonia thus acts like a sponge. It sops up the heat in one place and parts with it in another, the same ammonia constantly going backward and forward to fetch and discharge more heat. The complete cycle of operation comprises three parts: —

1st. *A compression side*, in which the gas is compressed.

2d. *A condensing side*, generally consisting of coils of pipe, in which the compressed gas circulates, parts with its heat and liquefies.

3d. *An expansion side*, consisting also of coils of pipe, in which the liquefied gas re-expands into a gas, absorbs heat, and performs the refrigerating work.

In order to render the operating continuous, these three sides or parts are connected together, the gas passing through them in the order named. The liquefied gas is allowed to flow into the expansion or evaporating coils, where it vaporizes and expands under a pressure varying from 10 to 30 pounds above that of the atmosphere, when ammonia is the agent in use. The gas then passes into the compressor, is compressed and forced into the condensers, where a pressure from 125 to 175 pounds per square inch usually exists; here liquefaction takes place and the resulting liquefied gas is allowed to flow to a stop-cock having a minute opening, which separates the compression from the expansion side of the plant. The expansion side consists of coils of pipe similar to those of the condensing side, but used for the reverse operation, which is the absorption of heat by the vaporization of liquefied gas instead of the expulsion of heat from it, as in the former operation. Heat is conducted through the expansion or cooling coils to, and is absorbed by, the vaporizing and expanding liquefied gas within such coils, for the reason that they are connected to the suction or low pressure side of the apparatus from which the compressors are continually drawing the gas and thereby reducing the pressure in said coils, as already stated, to a pressure of 10 to 30 pounds above the atmosphere; it being kept in mind that liquefied ammonia in again assuming a gaseous condition, has the power or capacity of reabsorbing, upon its expansion, a large quantity of heat. The liquefied gas entering these coils through the minute openings of the stop-cock, above referred to, is relieved of a pressure of 125 to 175 pounds, the amount requisite to maintain it in a liquid condition, when it begins to boil, and in so doing passes into the gaseous state. To do this it must have heat, which can be supplied only from the substance surrounding the pipes, such as air, brine, wort, etc. As a natural result the surrounding substances are reduced in temperature, or cooled.

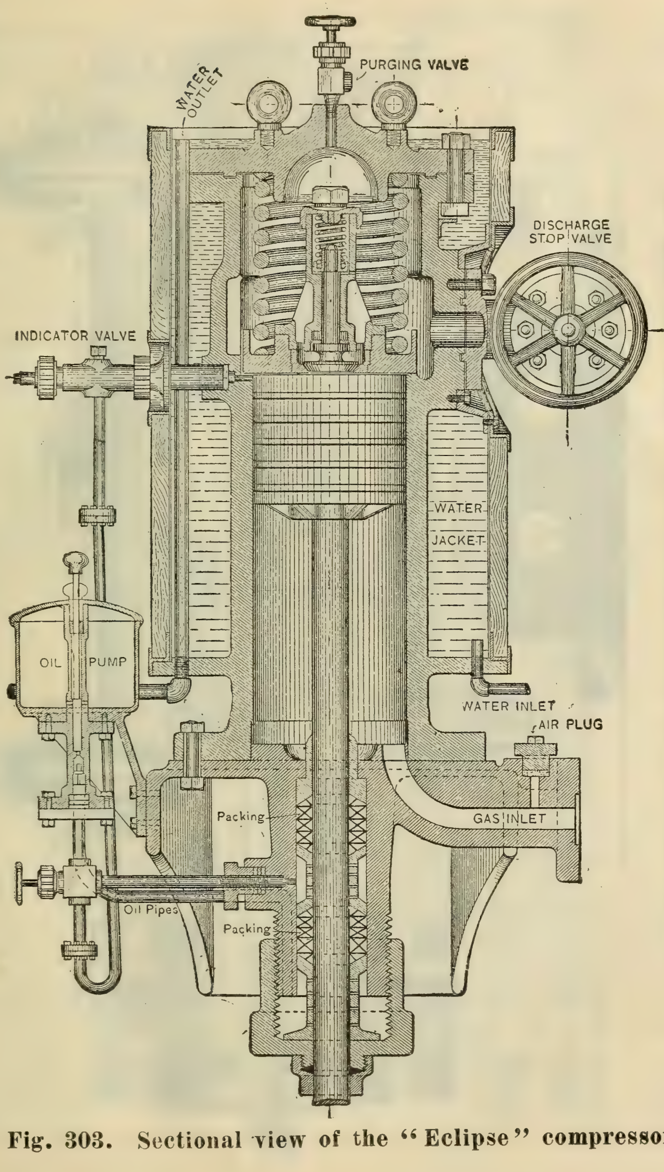


Fig. 303. Sectional view of the "Eclipse" compressor.

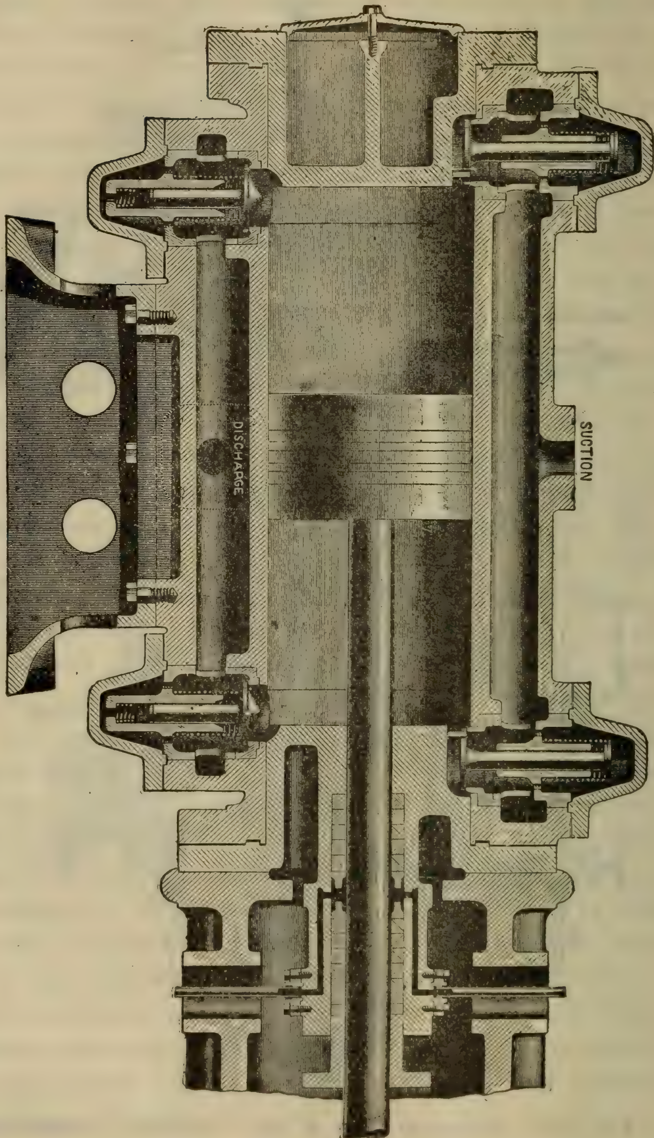


Fig. 304. Sectional view of double-acting horizontal compressor.

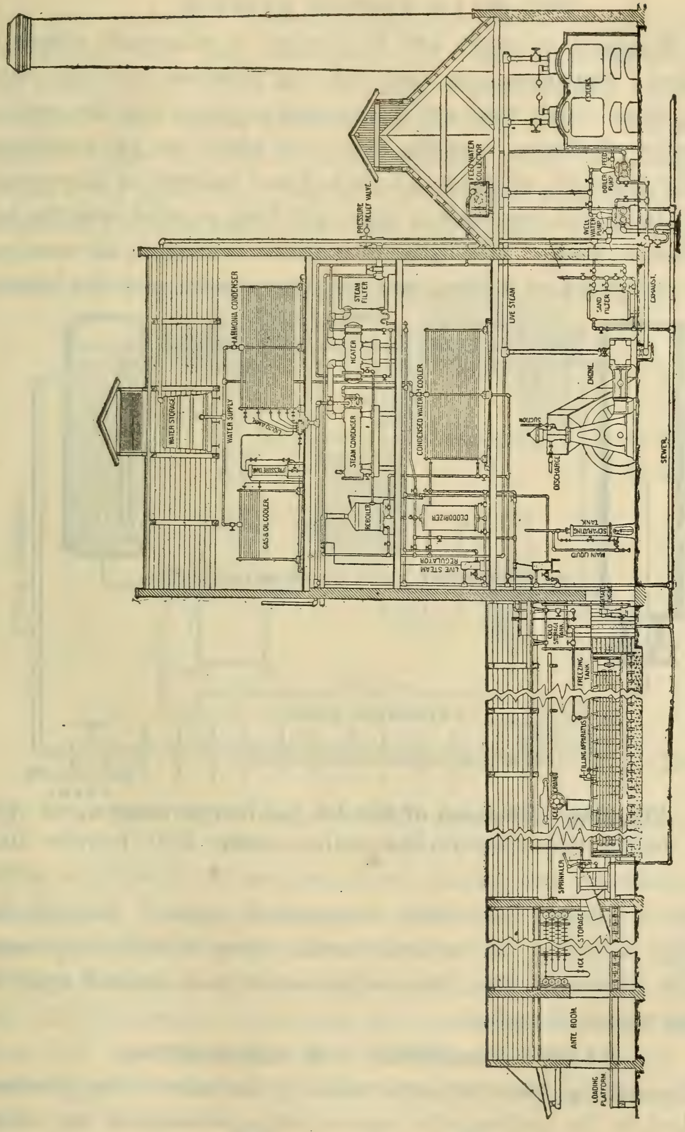


Fig. 305. De La Vergne ice-making plant, complete.

THE DE LA VERGNE SYSTEM.

The diagram on page 640 is seen to be extremely simple in conception; ammonia, gas and oil are received into the compressor, from which they are discharged together into the cooler. The cooled oil drops into the first tank while the gas continues into the condenser, where it is liquefied and collects in the second tank. The liquid ammonia is taken off from a point near the top of the second tank. If a little oil is taken over from the condenser it is conveyed by a pipe, as shown, to a point near the bottom

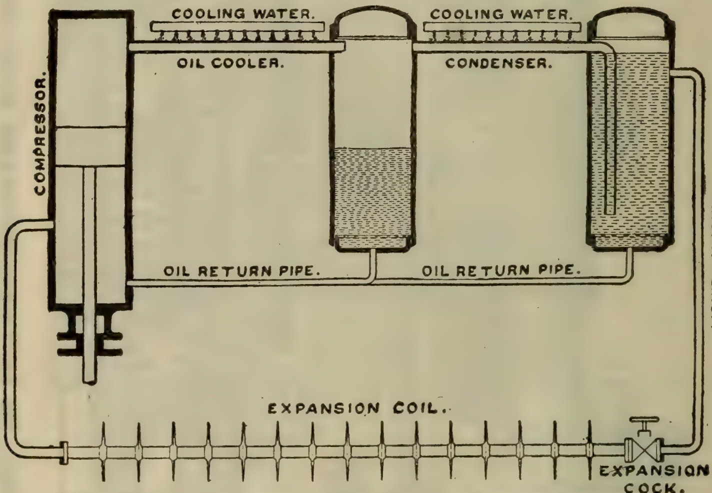


Fig. 306. Diagram of the De La Vergne system.

of the second tank, where it remains, since it is heavier than liquid ammonia, and cannot rise to get into the liquid pipe of the ammonia supply. The liquid ammonia is passed through the expansion cock into the expansion coil, where it boils into vapor which is drawn off into the compressor to pass around again in the order above described.

RATING MACHINES FOR ICE-MAKING.

Refrigerating machines are rated by the effect they produce equivalent to the melting of a corresponding amount of ice. Now the melting of one pound of ice is equivalent to the absorbing of

142 units of heat. In making ice from water, we have, however, to remove more than 142 units. We have first of all to reduce the water to 32° before we are ready to produce ice. If the water is at 82° this means the removal 50 heat units. Moreover, we cannot make ice with economy without going to a temperature much lower than 32°. The ice when formed may have a temperature of 18°, and the specific heat of ice being 0.5 this means the

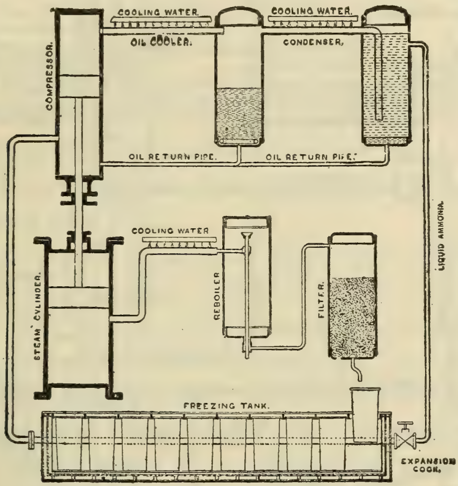


Fig. 307. Showing in diagrammatic form, the general outline of the process of ice-making with cans.

removal of 7 more heat units. In other words, we have to remove 199 heat units instead of 142 to produce a lb. of ice. Thus a 200-ton machine which would easily produce a refrigerating effect equal to the melting of 200 tons of ice would only produce 142 tons of actual ice. This proportion is still further reduced by the inevitable losses attending the use of large freezing tanks and the handling of the ice.

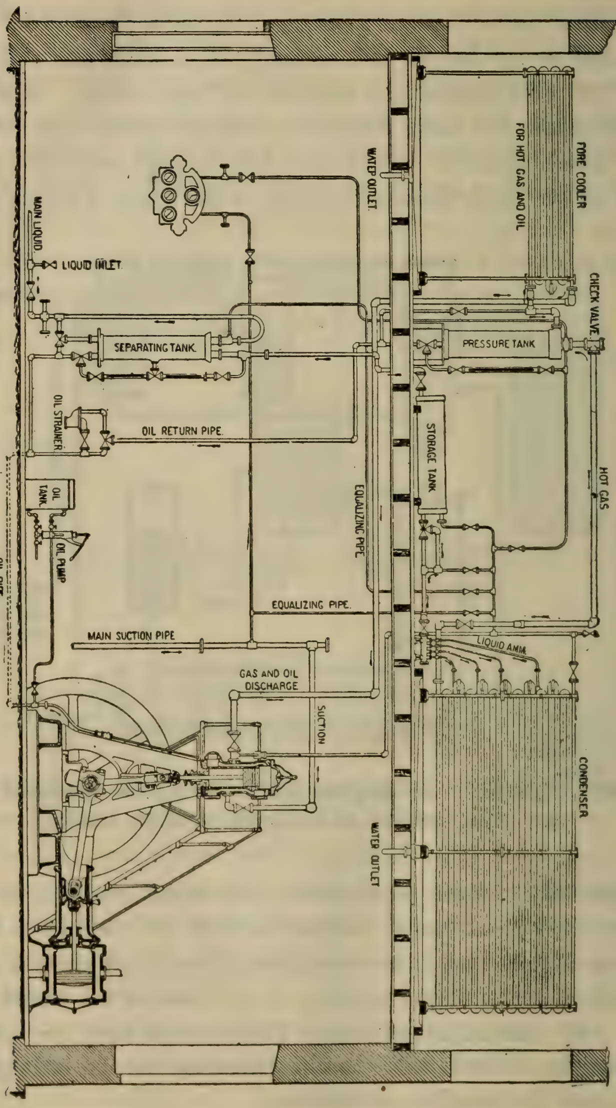


Fig. 308. A complete refrigerating plant, De La Vergue compressor.

COMPLETE CYCLE STANDARD DE LA VERGNE VERTICAL MACHINE.

The cut on preceding page shows the engine-room connections for the double acting vertical compressor complete with Corliss engine. The course of the gas can be very readily followed: After being discharged from the compressor it rises to the *fore cooler*, where the oil is cooled and deposited in the *pressure tank*. The ammonia gas goes on to the *condenser*, which it enters at the bottom. As fast as the liquid ammonia collects in the condenser, it is drawn off at different levels in the manner already described in connection with the condensers. From the storage tank it falls into the separating tank, where any remaining oil is trapped, and the anhydrous ammonia passes into the rooms to be cooled by way of the *main liquid pipe*.

The sectional view on following page, represents one of the De La Vergne double acting vertical compressors, as arranged for use with oil, as a sealing, lubricating and cooling agent. Two passages, marked " suction" and " discharge," respectively, connect the compressor with the pipe system. On the up stroke, gas flows through the lower suction valve into the space behind the moving piston, while the gas above the piston, after being compressed to the condenser pressure, is discharged through the upper valves (in the loose head) into the discharge passage. On the down stroke, gas flows into the cylinder through the upper suction valves, and the gas below the piston is compressed and passes through the lower discharge valves into the discharge passage. The piston in its downward course, closes successively the openings of these two discharge valves. When the lower is closed, however, the upper one communicates with the chamber in the piston, and the gas and oil still remaining below the piston are discharged through the valves into the chamber and out by the upper discharge valve. The oil being injected directly into

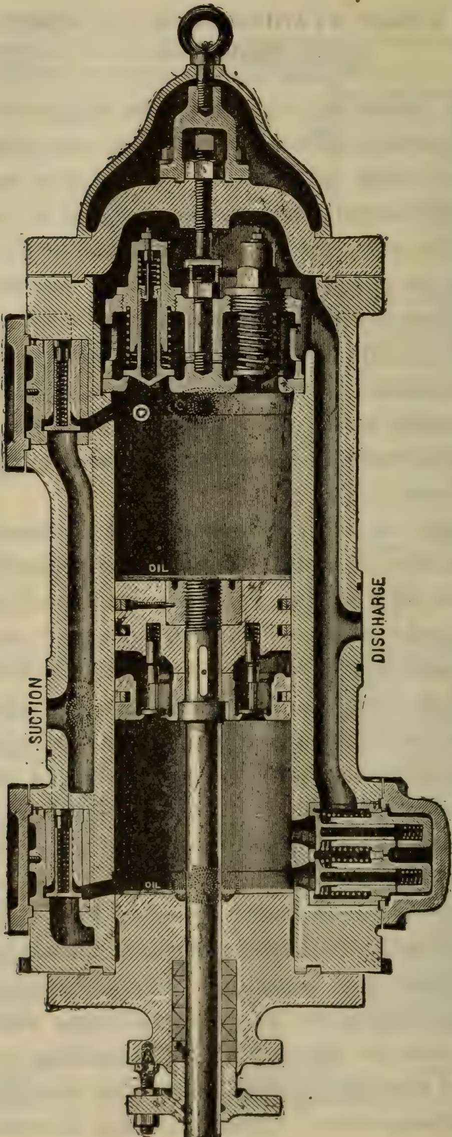


Fig. 309. Sectional view of De La Vergne double-acting vertical compressor.

the compressor after the compression of the full cylinder of gas has commenced, does not reduce the capacity of the machine.

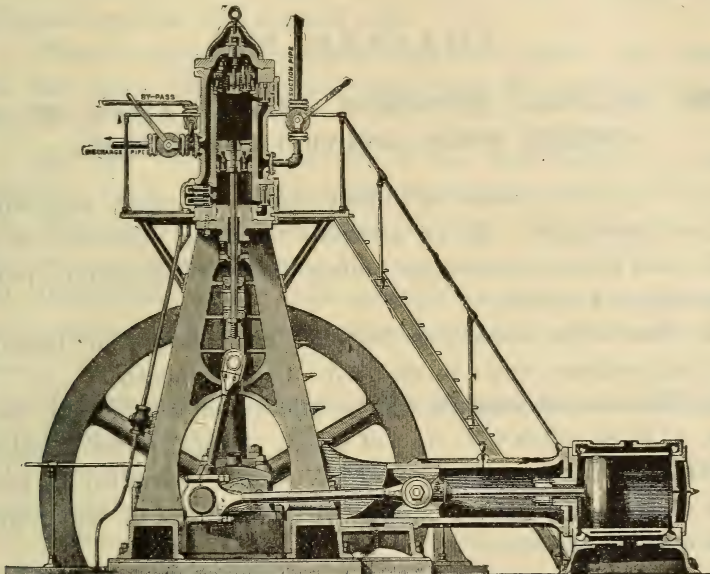


Fig. 310. De La Vergne double-acting compressor.

The above is a cut of the De La Vergne double acting compressor, driven by a Corliss engine. Both the compressor and engine cylinder, affording an opportunity of observing the relative positions of the pistons in each.

The oil for " cooling, sealing and lubricating " is brought to the compressor by the pipe running along the back of the " A " frame. The pipe marked " By-pass " is used when any portion of the pipe system in the engine house is to be independently exhausted of gas.

CHAPTER XXII.

SOME PRACTICAL QUESTIONS USUALLY ASKED OF ENGINEERS WHEN APPLYING FOR LICENSE.

Q. If you were called on to take charge of a plant, what would be your first duty? A. To ascertain the exact condition of the boiler and all its attachments (safety-valve, steam-gauge, pump, injector) and engine.

Q. How often would you blow off and clean your boilers if you had ordinary water to use? A. Twice a month.

Q. What steam pressure will be allowed on a boiler 50″ diameter, $\frac{3}{8}$″ thick, 60,000 T. S. $\frac{1}{6}$ of tensile strength factor of safety? A. One-sixth of tensile strength of plate, multiplied by thickness of plate, divided by one-half of the diameter of boiler, gives safe working pressure.

Q. How much heating surface is allowed per horse-power by builders of boilers? A. 12 to 15 feet for tubular and flue boilers.

Q. How do you estimate the strength of a boiler? A. By its diameter and thickness of metal.

Q. Which is the best, single or double riveting? A. Double riveting is from 16 to 20 per cent stronger than single.

Q. How much grate surface do boiler-makers allow per horse-power? A. About $\frac{2}{3}$ of a square foot.

Q. Of what use is a mud drum on a boiler, if any? A. For collecting all the sediment of a boiler.

Q. How often should it be blown out? A. Three or four times a day, in the morning before starting, and at noon.

Q. Of what use is a steam dome on a boiler? A. For storage of dry steam.

Q. What is the object of a safety-valve on a boiler? A. To relieve over pressure.

Q. What is your duty with reference to it? A. To raise it once a day and see that it is in good order.

Q. What is the use of a check valve on a boiler? A. To prevent the water from returning back into the pump or injector which feeds the boiler.

Q. Do you think a man-hole in the shell on top of a boiler weakens it any? A. Yes, to a certain extent.

Q. What effect has cold water on hot boiler plates? A. It will crack or fracture them.

Q. Where should the gauge cocks be located? A. The lowest gauge cock ought to be placed about 3 inches above the top row of flues.

Q. How would you have your blow-off located? A. In bottom of mud drum or boiler.

Q. How would you have your check valve arranged? A. With a stop cock between check and boiler.

Q. How many valves are there in a common plunger force pump? A. Two — a receiving and a discharge valve.

Q. How are they located? A. One on the suction side, the other on the discharge.

Q. How do you find the proper size of safety valves for boilers? A. Three square feet of grate surface is allowed for one inch area of spring-loaded valves, or two square feet of grate surface to one inch area of common lever valves.

Q. Give the reasons why pumps do not work sometimes? A. Leak in suction, leak around plunger, leaky check valve, or valves out of order, or lift too long.

Q. How often ought boilers to be thoroughly examined and tested? A. Twice a year.

Q. How would you test them? A. With hammer and with hydrostatic test — using warm water.

Q. Describe the single acting plunger pump; how it gets and discharges its water? A. The plunger displaces the air in the suction pipe, causing a vacuum, which is filled by the atmosphere forcing the water therein; the receiving valve closes and the plunger forces the water out through the discharge valve.

Q. What is the most economical boiler feeder? A. An exhaust injector.

Q. What economy is there in the exhaust injector? A. From 15 to 25 per cent saving in fuel.

Q. Where is the best place to enter the boiler with the feed water? A. Below the water level, but so that the cold water cannot strike hot plates. If injector is used this is not so material, as the feed water is always hot.

Q. What are the principal causes of priming in boilers? A. Too high water, not steam room enough, misconstruction, engine too large for boiler.

Q. How do you change the water in the boiler when steam is up? A. By putting on more feed and opening the surface skimmer or blow-off valve.

Q. If the safety valve was stuck how would you relieve the pressure on the boiler if the steam was up and could not make its escape? A. Work the steam off with engine after covering fires heavy with coal or ashes, and when the boiler is sufficiently cool, put safety valve in working order.

Q. If water in boiler is suffered to get low, what may be the result? A. Burn top of tubes, perhaps cause an explosion.

Q. If water is allowed to get too high, what result? A. Cause priming, perhaps cause breaking of cylinder head.

Q. What are the principal causes of foaming in boilers? A. Dirty and impure water and animal oil or grease.

Q. How can foaming in boilers be stopped? A. Close throttle and keep closed long enough to show true level of water. If that level is sufficiently high, feeding and blowing off will usually suffice to correct the evil.

Q. What would you do if you should find your water gone from sight very suddenly? A. If a light fire draw and cool off as quickly as possible; if a heavy fire cover with wet ashes or slack coal. Never open or close any outlets of steam when your water is out of sight.

Q. What precautions should you take to blow down a part of the water in your boiler while running with a good fire? A. Never leave the blow-off valve, and watch the water level.

Q, How much water would you blow off at once while running? A. Never blow off more than one gauge of water at a time while running.

Q. What precautions should the engineer take when necessary to stop with heavy fires? A. Close dampers, put on injector or pump, and if a bleeder is attached, use it.

Q. What is an engineer's first duty on entering a boiler-room? A. To ascertain the true water level, and look at steam gauge.

Q. When should a boiler be blown out? A. After it is cooled off — never while it is hot.

Q. When laying up a boiler what should be done? A. Clean thoroughly inside and out; remove all "Rust" and paint rust places with red lead; examine all stays and braces to see if any are loose or badly worn.

Q. Of what use is the indicator? A. The indicator is used to determine the power developed by an engine, to serve as a guide in setting valves and showing the action of steam in the cylinder.

Q. How would you increase the power of an engine? A. To increase the power of an engine, increase the speed, or get higher pressure of steam; or use less expansion.

Q. How do you find the horse-power of an engine?

$$\text{H. P.} = \frac{\text{area of piston} \times \text{M.E.P.} \times \text{piston speed.}}{33{,}000.}$$

Q. Which has the most friction, a perfectly fitted, or an imperfectly fitted valve or bearing? A. An imperfect one.

Q. How hot can you get water under atmospheric pressure with exhaust steam? A. 212°.

Q. Does pressure have any influence on the boiling point? A. Yes.

Q. Which do you think is the best economy, to run with your throttle wide open or partly shut? A. Always have the throttle wide open on a governor engine.

Q. At what temperature has iron the greatest tensile strength? A. About 600°.

Q. About how many pounds of water are required to yield one horse-power with our best engines? A. From 15 to 30.

Q. What is meant by atmospheric pressure? A. The weight of the atmosphere.

Q. What is the weight of atmosphere at sea level? A. 14.7 pounds per square inch.

Q. What is the coal consumption per hour per indicated horse-power? A. Varies from $1\frac{1}{2}$ to 7 lbs.

Q. What is the consumption of coal per hour on a square foot of grate surface? A. From 10 to 12 lbs.

Q. What is the water consumption in pounds per hour per indicated horse-power? A. From 15 to 45 lbs.

Q. How many pounds of water can be evaporated with one pound of best soft coal? A. From 7 to 10 lbs.

Q. How much steam will one cubic inch of water evaporate under atmospheric pressure? A. One cubic foot of steam (approximately).

Q. What is the weight of a cubic foot of fresh water? A. 62.425 lbs.

Q. What is the weight of a cubic foot of wrought iron? A. 480 lbs.

Q. What is the last thing to do at night before leaving the plant? A. Look around for greasy waste, hot coals, matches, or anything which could fire the building.

Q. What is the weight of a square foot of one-half inch boiler plate? A. 20 lbs.

Q. How much wood equals one ton of soft coal for steam purposes? A. About 4,000 lbs. of wood.

Q. What is the source of all power in the steam engine? A. The heat stored up in the coal.

Q. How is the heat liberated from the coal? A. By burning it — that is, by combustion.

Q. Of what does coal consist? A. Carbon, hydrogen, nitrogen, sulphur, oxygen and ash.

Q. What are the relative proportions of these that enter into coal? A. There are different proportions in different specimens of coal, but the following shows the average per cent: Carbon, 80; hydrogen, 5; nitrogen, 1; sulphur, 2; oxygen, 7; ash, 5.

Q. What must be mixed with coal before it will burn? A. Air.

Q. Of what is air composed? A. It is composed of nitrogen and oxygen in the proportion of 77 per cent nitrogen to 23 of oxygen.

Q. What parts of the air mix with what parts of coal? A. The oxygen of the air mixes with the carbon and hydrogen of the coal.

Q. How much air must mix with coal? A. 300 cubic feet of air for every pound of coal.

Q. How many pounds of air are required to burn one pound of carbon? A. From 20 to 24, generally taken at 24.

Q. How many pounds of air to burn one pound of hydrogen? A. Thirty-six.

Q. Is hydrogen hotter than carbon? A. Yes, $4\frac{1}{4}$ times hotter.

Q. What part of the coal gives out the most heat? A. The hydrogen does part for part, but as there is so much more of carbon than hydrogen in the coal, we get the greatest amount of heat from the carbon.

Q. In how many different ways is heat transmitted? A. Three, by radiation, by conduction and convection.

Q. If the fire consisted of glowing fuel, show how the heat enters the water and forms steam? A. The heat from the glowing fuel passes by radiation through the air space above the fuel to the furnace crown; there it passes through the iron of the crown by conduction; there, it warms the water resting on the crown, which then rises and parts with its heat to the colder water by conduction till the whole mass of water is heated; then the heated water rises to the surface and parts with its steam, so a constant circulation is maintained by convection,

Q. Of what does water consist? A. Oxygen and hydrogen.

Q. In what proportion? A. Eight of oxygen to one of hydrogen, by weight.

Q. What are the different kinds of heat? A. Latent heat, sensible heat and sometimes, total heat.

Q. What is meant by latent heat? A. Heat that does not affect the thermometer and which expends itself in changing the nature of a body, such as turning ice into water or water into steam.

Q. Under what circumstances do bodies get latent heat? A. When they are passing from a solid state to a liquid state, or from a liquid to a gaseous state.

Q. How can latent heat be recovered? A. By bringing the body back from a state of gas to a liquid, or from that of a liquid to that of a solid.

Q. What is meant by a thermal unit? A. The heat necessary to raise one pound of water, at any temperature—one degree Fah.

Q. If the power is in coal, why should we use steam? A. Because, steam has some properties which make it an invaluable agent for applying the energy of the heat to the engine.

Q. What is steam? A. It is an invisible elastic gas generated from water by the application of heat.

Q. What are the properties which make it so valuable to us?

A. 1. The ease with which we can condense it. 2. Its great expansive power. 3. The small space it occupies when condensed.

Q. Why do you condense the steam? A. To form a vacuum and so destroy the back pressure that would otherwise be on the piston, and thus get more useful work out of the steam.

Q. What is vacuum? A. A space void of pressure.

Q. How do you maintain a vacuum? A. By the steam used being constantly condensed by the cold water or cold tubes, and the air pump constantly clearing the condenser of air.

Q. Why does condensing the used steam form a vacuum? A. Because a cubic foot of steam at atmospheric pressure shrinks into about a cubic inch of water.

Q. What do you understand by the term horse-power? A. A horse-power is equivalent to raising 33,000 lbs. one foot per minute, or 550 lbs. raised one foot per second.

Q. What do you understand by lead on an engine's valve? A. Lead on a valve is the admission of steam into the cylinder before the piston starts its stroke.

Q. What is the clearance of a cylinder as the term is applied at the present time? A. Clearance is the space between the cylinder head and the piston head, with ports included.

Q. What are considered the greatest improvements on the stationary engine in the last forty years? A. The governor, the Corliss valve gear, and the triple expansion engine.

Q. What is meant by triple expansion engine? A. A triple expansion engine has three cylinders, using the steam expansively in each one.

Q. Is there any danger of a well-fitted and tightly-keyed fly-wheel coming loose? A. Yes; water in the cylinder by producing a heavy jar would tend to loosen a fly-wheel and frequently reversing an engine under a load and high speed, would tend to produce the same effect.

Q. What is a condenser as applied to an engine? A. The condenser is a part of the low-pressure engine, and is a receptacle into which the exhaust enters and is there condensed.

Q. What are the principles which distinguish a high-pressure from a low-pressure engine? A. Where no condenser is used and the exhaust steam is open to the atmosphere it is high pressure.

Q. About how much gain is there by using the condenser? A. 17 to 25 per cent, where cost of water is not figured.

Q. What do you understand by the use of steam expansively? A. Where steam admitted at a certain pressure is cut off and allowed to expand to a lower pressure.

Q. How many inches of vacuum give the best results in a condensing engine? A. Usually considered 25″.

Q. What is meant by a horizontal tandem engine? A. One cylinder being behind the other, with two pistons on same rod.

Q. What is a Corliss valve gear? A. (Describe the half moon, or crab-claw gear, or oval-arm gear with dash pots.)

Q. From what cause do belts have the power to drive shafting? A. By friction or adhesion.

Q. What do you understand by lap? A. Outside lap is that portion of valve which extends beyond the ports when valve is placed on the center of travel; and inside lap is that portion of valves which projects over the ports on the inside or towards the middle of valve.

Q. What is the use of inside lap? A. To give the engine compression.

Q. Where is the dead center of an engine? A. The point where the crank and the piston rod are in the same right line.

Q. In what position would you place an engine to take up any lost motion of the reciprocating parts? A. Place the engine in the position where the least wear takes place on the journals. That is, in taking up the wear of crank-pin brasses, place the engine on either dead center, as when running, there is little wear

upon the crank-pin at these points. If taking up the cross-head pin brasses — without disconnecting and swinging the rod — place the engine at half stroke, which is the extreme point of swing of the rod, there being the least wear on the brasses and cross-head pin in this position.

Q. What benefits are derived from using fly-wheels on steam engines? A. The energy developed in the cylinder while the steam is doing its work, is stored up in the fly-wheel, and given out by it while there is no work being done in the cylinder — that is, when the engine is passing the dead centers. This tends to keep the speed of the engine shaft steady.

Q. Name several kinds of reducing motions, as used in indicator practice? A. The pantograph, the pendulum, the brumbo pulley, the reducing wheel.

Q. How can an engineer tell from an indicator diagram whether the piston or valves are leaking? A. Leaky steam valves will cause the expansion curve to become convex; that is, it will not follow hyperbolic expansion, and will also show increased back pressure. But if the exhaust valves leak also, one may offset the other, and the indicator diagram would show no leak. A leaky piston can be detected by a rapid falling in the pressure on the expansion curve immediately after the point of cut-off. It will also show increased back pressure. A falling in pressure in the upper portion of the compression curve shows a leak in the exhaust valve.

Q. What would be the best method of treating a badly scaled boiler, that was to be cleaned by a liberal use of compound? A. First, open the boiler up and note where the loose scale, if any, has lodged. Wash out thoroughly and put in the required amount of compound. While the boiler is in service, open the blow-off valve for a few seconds, two or three times a day, to be assured that it does not become stopped up with scale. After running the boiler for a week, shut it down, and when the

pressure is down and the boiler cooled off, run the water out and take off the hand-hole plates. Note what affect the compound has had on the scale, and where the disengaged scale has lodged. Wash out thoroughly and use judgment as to whether it is advisable to use a less or greater quantity of compound, or to add a small quantity daily. Continue the washing out at short intervals, as many boilers have been burned by large quantities of scale dropping on the fire sheets and not being removed.

Q. What is an engineer's first duty upon taking charge of a steam plant? A. The first duty of an engineer assuming charge of a steam plant is to familiarize himself with his surroundings, ascertain the duty required of each and every piece of machinery contained therein, and in just what condition each one is. Let us discuss it at length, assuming that when just engaged he is informed as to the nature of the work required of the plant in question, namely: Whether it is a heating plant, electric lighting, hydraulic or electric elevator, power station, or any other kind of the various steam plants in existence. Of course, a great deal depends upon the size and kind of plant under consideration and the number of men employed, hours in operation, and some other things in general which most engineers know of. He should first see just what his plant contains "from cellar to garret," so to speak; whether all that is contained has to run continually, or almost so, and what can be depended on in case anything should suddenly become deranged or give out entirely. Next, he should ascertain the general condition of everything, going over each portion in turn, as time and opportunity permit, and conclude from what he has seen how much longer it may be run safely and economically. It will be remembered that a piece of machinery may be run safely and yet not with economy. So, if he should wait for the safety limit to be reached, without taking other things into consideration, he might wait

a long time and in so doing waste many dollars of his employer's money before it was thought necessary to renovate, repair or renew. In going over everything, examining each part critically, it would be well to make copious notes, and, sketches might be added, to which the engineer can again refer. It sometimes happens that engineers, in making an examination of machinery, do not take dimensions or make sketches of certain parts, which have to be repaired, or perhaps renewed, thinking that the next time the apparatus is looked at will do for that. Now, it sometimes happens that the "next time" is the time when some accident occurs, finding him unprepared, causing confusion, in the midst of which the making of sketches and taking of dimensions cannot be thought of. All such should be done at the first opportunity, and spare parts of the different machinery should be kept on hand, especially in the case of a plant which has only the machinery which is constantly in use. Another point of importance to which an engineer should give attention, is to ascertain the quantity and kind of supplies which are on hand, that he may know when to make requisition for more, and so not run short, as he otherwise might do. It is also important to see what tools the plant contains and upon what he can depend in case of the break-down of any part of the machinery. Of course all the above cannot be done in one day, but no time should be lost in doing all these things as early as possible, for the sooner he gets all the particulars and details of his plant at his "fingers' ends," the lighter will be his own labors, and the more free will his mind be to think and act intelligently for the emergencies of the future. Therefore, by performing this first duty as early and thoroughly as possible, the succeeding ones will be comparatively easy to handle and perform, for the reason that he will be prepared for them.

Q. Define and explain the difference between sensible and latent heat? A. The difference between sensible and latent heat

is explained thus: Sensible heat may be measured with a thermometer, that is, it affects the mercury in a thermometer, causing it to rise in the stem so that the degree of heat may be measured on the graduated scale affixed. Latent heat does not affect the thermometer. Bodies get latent heat when they are passing from a solid state to a liquid state, and also when passing from a liquid to a gaseous state; and moreover, this latent heat can be recovered by bringing a body back from a gaseous to a liquid state, and from liquid to solid. Water is most commonly seen under the three forms of matter just mentioned, namely, solid, ice; liquid, water; gaseous, steam. The following method has been used to explain how latent heat exists: A quantity of powdered ice is placed in a vessel and brought into a very warm room. As long as it remains as ice, it may be any degree of heat below 32° Fahr., but the instant it begins to melt, owing to the heat of the room, a thermometer placed in it will record 32° Fahr. The thermometer will continue at 32° as long as there is any ice in the vessel, but just as soon as the last piece of ice has melted it will begin to rise, and continue to do so until the water boils, when it will stand at 212° ; but although the water goes on receiving heat after this, the instrument will stand at 212° until all the water has boiled away. Now, a great amount of heat must have entered the water since the ice began to melt, but it has no effect on the thermometer, which continues at 32°, as noted above; the heat that has so entered is called "the latent heat of water." The heat that has entered the water from boiling till it all becomes steam is called the "latent heat of steam." The latent heat of water has been found to be 143° Fahr. and the latent heat of steam, at the pressure of the atmosphere, is 966°. This is the way the above was determined: A quantity of water at a temperature of 32° Fahr. is made to boil, and the time taken to do so noted; in this case, it took one hour. The water must be kept boiling until it has all evaporated,

and the time noted from boiling till evaporation, which in this case will be $5\frac{1}{3}$ hours. Therefore,

Temperature of boiling point, ...	212°
Temperature of water at first, ...	32°
Heat that has entered the water in one hour, ...	180°
Number of hours boiling, ...	$5\frac{1}{3}$
	900
	60
Heat that has entered during the $5\frac{1}{3}$ hours, ...	960°

From this we see that the heat necessary to form steam, instead of being only 212°, must be $966° + 212° = 1178°$, or $5\frac{1}{2}$ times as great. Therefore, if it were not for latent heat, we would require to burn $5\frac{1}{2}$ times the amount of coal that we now do to generate steam. The sensible and latent heats alter with the pressure, but as the sensible increases the latent decreases, and, roughly speaking, the total heat, or the sum of the two, is the same. In connection with the foregoing questions, we would recommend the reader to spend a little time in looking over the "steam tables," and make comparisons between the different quantities noted therein. By so doing he will get an exact knowledge of the properties of saturated steam.

Q. Explain the term "clearance," as used in connection with an engine cylinder? A. There are two kinds of clearance, cylinder clearance and piston clearance. Cylinder clearance means the space or volume, which exists between the piston and the valve, when the piston is exactly at the beginning of the stroke and the crank is on the dead center. This volume can be found by taking careful and exact measurements and making calculations from them, but a more correct way is to fill the space with water, noting the quantity used, and so make calculations to find the cubic con-

tents. The cubic contents of the clearance space is a certain percentage of the total volume of the cylinder itself and such clearance is expressed as so much per cent. This clearance causes a small loss of steam each stroke, owing to the difference between the initial and compressive pressure. Piston clearance is the space between the piston and cylinder head when the crank is on the dead center. This clearance is necessary to prevent the cylinder head being knocked out, in case of an unusual quantity of water gaining entrance to the cylinder while the engine is running at its usual speed; and also to admit of the crank-pin and wrist-pin brasses being keyed up at certain intervals. The way to find the piston clearance of an engine is as follows: First, disconnect the wrist-pin end of the connecting rod from the cross-head, and with a bar push back the cross-head until the piston strikes the cylinder head; then make a mark with a scriber or sharp chisel, on both the sides of the cross-head and on the guide in which the cross-head runs; these marks must be exactly in line with each other while the piston is in the above stated position. Next, move the piston to the other end of the cylinder till it strikes the head, and make a mark on the guide similar to that on the other end, using the same mark which was made on the cross-head. The new mark must also be in line with this, as at the first mentioned end. We now have a mark at each end of the guide, which represents the place at which the piston strikes the cylinder head, when they alternately coincide with the mark on the cross-head itself. Now, connect the rod to the cross-head again and place the engine or crank on the center. Next, produce or extend the mark on the cross-head to the guide, this time using a pencil instead of a chisel and scriber. The distance between the new pencil mark and the first mark made on the guide is the amount of piston clearance which exists at that end of the cylinder. Repeat the operation on the other end and we will obtain the clearance existing there. If these clear-

ances are not equal, as indicated by the marks, make them so by the means provided for in the design of the piston rod and crosshead. After the clearance has been equalized, the pencil marks may be obliterated and marks similar to the first ones may be cut in, thus leaving a permanent mark, which can be seen while the engine is running, and from which can be determined whether the clearance is lessening, and at which end.

Q. What is the pressure of the atmosphere at the sea level, and how determined? A. The pressure of the atmosphere is generally spoken of as 15 lbs. per square inch, but as the pressure of the atmosphere is constantly varying at any one spot, corrections have to be made according to the reading of a barometer. Generally speaking, 15 is as nearly correct as engineers require it. The pressure of the atmosphere can be ascertained by the following experiment: Take a glass tube about 33 inches long, having a bore equal to a square inch in section. Let one end of the tube be closed in or capped, so that it can contain a fluid. Then fill it with pure mercury, carefully expelling any air bubbles. When it is full, cover the open end of the tube with a piece of glass and invert the whole tube. Place the open end into a cup of mercury, the surface of which is subject to the pressure of the air, and then withdraw the piece of glass. The mercury in the tube will drop about three inches and then stop. When it has ceased to fall, again cover the end of the tube with the glass. Lift the tube out of the cup and remove the glass so that the mercury may run out into a scale-pan provided for that purpose. Upon actually weighing the mercury lately contained in the tube, it will be found to weigh 14.7 lbs. The mercury will stop falling in the tube at 30 inches, or at the sea level. Hence, we know that the atmosphere balances, or exerts a pressure of 14.7 lbs. per square inch at the sea level.

Q. Upon what does the efficiency of a surface condenser depend? A. The efficiency of a surface condenser depends upon:

1st, the proper amount of cooling surface; 2d, the rapidity with which the water is made to circulate through the tubes; 3d, the water being made to flow in an opposite direction to the steam. The temperature of the circulating water also has a bearing on the question, as it is obvious that the colder the water the more effective it will be in condensing the steam.

Q. A feed pump has a steam cylinder of 6 inches in diameter, and water cylinder of 4 inches diameter; assuming the steam pressure carried to be 80 lbs. per square inch throughout the stroke, what will be the balancing pressure per square inch against the water piston, friction being entirely neglected, and gauge pressure being used? A. In this question, we first find the area, the number of square inches contained in the steam piston. Thus: The diameter = 6 in. and $6^2 \times .7854$ = the area. Worked out it appears thus: 6^2 means that 6 is to be squared, or multiplied by itself, or $6 \times 6 = 36$ square inches, and 36 square inches multiplied by the constant $.7854 =$ 28.27 square inches area contained in the steam piston. Since the pressure is stated to be 80 lbs. per square inch, then $28.27 \times 80 =$ total pressure on the piston in pounds, or 2261.60 lbs. Now, we will find the area of the water piston, which is 4 inches in diameter, $4^2 \times .7854 = 12.5664$ square inches contained in the water piston. Therefore, the water piston, with an area of 12.56 sq. in., has to have a resistance against which it will act of 2261.60 lbs., in order to balance the pressure against the steam piston. Hence, the pressure per square inch can be found by dividing 2261.60, or 2261.60 divided by 12.56 = 180 lbs. per square inch, the balancing pressure on the 4-inch water piston.

Q. State what you consider a good standard of strength for steel boiler plate? A. The American Boiler Makers' standard, as used, is as follows: Tensile strength, from 55,000 to 60,000 lbs. per square inch section; elongation in 8 inches, 20 per cent for plates $\frac{3}{8}$ inch thick and under; 22 per cent for plates $\frac{3}{8}$ to $\frac{3}{4}$

inches; 25 per cent for plates $\frac{3}{4}$ inch and under; the specimen test piece must bend back on itself when cold, without showing signs of fracture; for plates over $\frac{1}{2}$ inch thick, specimens must withstand bending 180° (or half way) round a mandrel $1\frac{1}{2}$ times the thickness of the plate. The chemical requirements are as follows: Phosphorus, not over .04 per cent; sulphur, not over .03 per cent.

Q. What is meant by the heating surface of a boiler? A. The heating surface of a boiler is that surface of plates or tubes on one side of which is water, and on the other, hot gases. It has been decided that the surface next the water shall be reckoned, the value to be given in square feet. In a fire tube, or tubular boiler, it will include the under side of the shell from fire-line to fire-line (usually about one-half of it), the tubes and such part of the back-tube sheet as is below the back arch and not taken up by the tube ends. For a water-tube boiler, the heating surface will include the tubes, such part of the headers as are in contact with the hot gases, and the lower part (about one-half) of the steam drum. In calculating the heating surface, none should be taken which has steam on one side and hot gases on the other, as such parts tend to superheat the steam, and are known as superheating surfaces.

Q. What is a boiler horse-power? A. A boiler horse-power has been recently defined as the evaporation of $34\frac{1}{2}$ pounds of water per hour from a feed water temperature of 212° Fahr. into steam at a temperature of 212° Fahr., and at a pressure of one atmosphere. Under these conditions each pound of water evaporated will take up 966 heat units, and the $34\frac{1}{2}$ lbs. will take $34\frac{1}{2} \times 966 = 33{,}327$ heat units per hour. Hence, to find the horse-power of a boiler, it is necessary to find the heat units delivered per hour to the water and divide that number by 33,327.

Q. What will be the heating surface of a fire-tube boiler 6 feet in diameter, having 150 tubes 3 inches in diameter and 15

feet long? A. Each tube will have a heating surface equal to its outside area, since the water is on the outside of the tubes. The area of a cylinder 3 in. in diameter and 15 ft. long will be the circumference times the length; 3 in. $= \frac{1}{4}$ ft. and the circumference $= 3.1416 \times \frac{1}{4} = .7854$ ft.; this, times the length 15 ft. $= 11.78$ sq. ft. for one tube; for 150 tubes, it will be 150 times that $= 1767$ sq. ft. The lower half of the shell is usually considered as heating surface. The circumference of a circle 6 ft. in diameter is $6 \times 3.1416 = 18.85$ ft. and the area of the shell $= 18.85 \times 15 = 282.75$ sq. ft. Half this will be 141.37 sq. ft. For the back end or tube plate, the total area will be the diameter squared times $.7854 = 6^2 \times .7854 = 28.27$ sq. ft.; $\frac{2}{3}$ of this will be below the arch, and $\frac{2}{3}$ of $28.27 = 18.85$ sq. ft. From this must be subtracted the area of the ends of the tubes. The end area of one tube is $(\frac{1}{4})^2 \times .7854 = .049$ sq. ft., and for 150 tubes it is 150 times that, or 7.35 sq. ft. The heating surface of the tube plate will then be 18.85 minus $7.35 = 11.5$ sq. ft. The front tube plate is not considered, because the gases are cooled too much to be effective by the time they have passed through the tubes. The total heating surface is $1767 + 141.37 + 11.5 = 1919.87$ sq. ft.

Q. On what does the efficiency of a boiler depend? A. The efficiency of any piece of machinery is the ratio of the energy made useful to that furnished. The object of the boiler is to make steam; hence, the enegy *used* is that which has gone into the steam. The proportion of the heat generated in the furnace which is transferred to the steam, will depend on the thickness of the plates of the boiler, on their condition as to cleanliness, on the amount of time during which the gases are in contact with the plates in their passage from furnace to chimney, on the completeness with which all parts of the gases are brought in contact with the plates, and on the temperature of the hot gases. Evidently, heat will pass through a thin plate more readily than through a thick one, and more

readily through a clean plate than through one on which a non-conducting coating of soot or scale has formed; the more time available for the transfer of heat, the greater will be the amount transferred; the more complete the contact between plates and gases, the more opportunity will there be for the transfer of heat, and the higher the temperature of the gases, the more rapidly will the heat be transferred. To have a boiler efficient, it is necessary to have plenty of heating surface, so that the hot gases will have time for contact, to keep the plates clean, to have good circulation of the gases, and to keep their temperature high by preventing radiation and allowing as little air to enter the furnace as is needed for good combustion. The efficiency of the furnace, that is, the ratio of the heat generated in the furnace to that contained in the coal, is a separate matter, though often the two are lumped together. It depends on the adaptation of the furnace to the kind and size of coal used, on the size of the combustion chamber and on the proper firing of the coal.

Q. On what its satisfactory working? A. In order to work satisfactorily, a boiler must not only be efficient, but must make steam rapidly, must make dry steam, must be easily fired and cleaned, and must be capable of standing a considerable amount of forcing without serious priming. To get rapid steam making, it is necessary to have good circulation of the water in the boiler; to get dry steam, plenty of steam space is needed, so that the steam may circulate slowly and allow the water to drop out of it; easy firing means a low fire door of good size, and a rather short grate; easy cleaning means accessible parts, good sized man-holes, good sized and well placed hand-holes, a large blow-off and a short boiler; the prevention of priming when carrying an overload is a difficult matter; the tendency to such an occurrence depends largely on the feed-water used; plenty of steam space and good circulation are helpful, but some waters will foam in spite of all precautions.

Q. Suppose a slide valve cutting off at $\frac{3}{4}$ stroke, and a $\frac{5}{8}$ cut-off is desired, how would you proceed? A. Put on a new valve with more outside lap. This would require a greater travel of the valve, and therefore would increase the throw of the eccentric, also.

Q. Which requires the greater outside lap, cutting off at $\frac{9}{16}$ of the stroke, or cutting off at $\frac{7}{8}$? A. Cutting off at $\frac{9}{16}$. The earlier the cut-off, the greater should be the outside lap.

Q. Are all plain slide valves made alike, as regards the exhaust cavity of the valve? A. No; sometimes they are made "line and line" inside, that is, the width of the exhaust cavity is equal to the distance between the inner edges of the two steam ports; and again, the width of the exhaust cavity is made greater or less than this distance, according as an earlier or later release is desired.

Q. What is the effect of giving inside lap to a slide valve? A. It delays the release and increases the compression.

Q. What is the effect of giving inside lead to a slide valve? A. It gives an early release and decreases the compression.

Q. Suppose a simple slide valve engine with a fly-ball governor, and the governor belt should break or slip off, what would happen? A. If it were a plain governor the engine would race; but if a governor with an automatic stop, the engine would slow down and stop.

Q. What two forces are opposed to each other in a case of fly-ball governor? A. Centrifugal force, tending to throw the balls away from the governor staff, and the force of gravity, tending to draw the balls to the staff.

Q. What other name is given to a fly-ball governor? A. It is also called a throttling governor, because the steam in passing through the governor valve is throttled, choked, or wire-drawn.

Q. Are all fly-ball governors throttling governors? A. No; the governor of a Porter-Allen engine and those of all Corliss

engines, while of the fly-ball type, are not throttling governors, because the steam does not pass through them.

Q. If the governor shaft of a fly-ball governor on a plain slide-valve engine should break, could the engine be run? A. Yes; by regulating the speed of the engine by hand at the throttle-valve.

Q. Describe an automatic cut-off engine? A. In this class of engines, as the load on the engine becomes greater or less, the steam entering the cylinder is cut off later or earlier, and it is done through a fly-ball governor in the case of a Corliss engine, or through a shaft-governor or regulator in the case of a high-speed engine.

Q. In an automatic cut-off high-speed engine with shaft-governor, is the eccentric fastened to the shaft? A. It is not. It is so arranged as to move freely across the shaft, in order to permit the center of the eccentric to approach or to recede from the center of the shaft, according as the load on the engine decreases or is increased. And herein lies the chief difference between a plain slide-valve and an automatic cut-off slide-valve engine.

Q. If the connecting rod of an engine had box liners at both ends and in taking it down the liners were all mixed up, how could the length of the rod from center to center of boxes be found? A. Put the cross-head in the middle of its stroke — after finding the piston striking points — and then measure from the center of the cross-head wrist to the center of the main shaft. If the piston clearance at both ends of the cylinder is known, the piston may be pushed to the crank end of the cylinder until it touches the head, and the distance from the center of the cross-head wrist to the center of the main shaft found, to which should be added the length or throw of the crank, and also the piston clearance at the crank end of the cylinder.

Q. But suppose it were more convenient to push the piston to the head end of the cylinder, what then? A. Find the distance

from the center of cross-head wrist to center of main shaft and deduct the throw of the crank, and also the clearance.

Q. How is the length of the valve stem and of the eccentric rod found for a plain slide valve engine having a rock shaft? A. If the motion of the slide valve is parallel with the motion of the piston, the length of the valve stem may be found by measuring in a horizontal line from the center of the valve seat to the center of the rock shaft; and for the eccentric rod by measuring from the center of the rock shaft, horizontally, to the center of the main shaft, which would include one-half the yoke.

Q. What is a direct, and also an indirect valve motion? A. When there is no rock shaft between the eccentric and the valve to compound the motion, it is called "direct," and when a rock shaft intervenes, it is called an "indirect" valve motion.

Q. Is the valve motion of a Corliss engine direct or indirect? A. It is direct.

Q. How so; it has a rock shaft between the eccentric and the wrist plate? A. Even so, it is a direct valve motion; because all connections to the rock-shaft arm are above the center of the shaft, consequently, the motion is simple and not compound.

Q. When is an engine said to "run under," and when to "run over?" A. When the crank pin is above the center of the main shaft and the pin moves towards the cylinder, the engine is said to "run under;" and when it moves away from the cylinder, the engine is said to "run over."

Q. What is meant by lead of valve, and what is it for? A. Lead is the amount that the port is open to steam when the crank is on its center. It is given in order to allow the full pressure of steam to come on the piston at the beginning of the stroke, and to provide a cushion for the piston.

Q. Could not cushion for the piston be obtained in some other manner? A. Yes, by producing compression by an early closing of the exhaust.

Q. Suppose a slide valve had $\frac{3}{4}''$ lap and no lead, and it was desired to give it $\frac{1}{32}''$ lead, how should it be done? A. By moving the eccentric.

Q. Why could it not be done by altering the length of the eccentric rod? A. Because the eccentric rod does not establish the amount of lead; it simply equalizes the lead given by moving eccentric.

Q. How would you test the piston of a steam engine to see whether it was steam-tight or not? A. Put the crank on the outboard center; remove the cylinder head on the head end; block the cross-head and admit steam to the crank-end of cylinder and note the effect. The fly-wheel, or the cross-head may be securely blocked and the piston tested in this manner at different points in the stroke.

Q. Why are two eccentrics and two wrist plates put on some Corliss engines? A. One eccentric is for the induction valves to lengthen the range of the cut-off; the other for the exhaust valves to admit of early release, without excessive compression. With a Corliss engine having but one eccentric, the limit of cut-off is at less than one-half stroke, but with two eccentrics the cut-off may be still later in the stroke, and still release the steam at the proper time.

Q. What is meant by a "blocked up" governor on a Corliss engine? A. When the safety stop is "in," the governor is said to be blocked up.

Q. With a blocked up governor, suppose the main driving belt should break, what would be the result? A. The engine would race and would, perhaps, be wrecked.

Q. What is meant by the fire line of a horizontal cylindrical boiler? A. It is the height to which the shell is exposed to the action of the flames.

Q. How high should the fire line be run? A. It may be run as high as the lower gauge cock water level, although it is frequently run no higher than the top row of flues.

Q. What causes a chimney or smoke-stack to draw? A. The difference in the temperature of the air inside the chimney and that outside. The air inside expands and exerts less pressure than the outside air, which rushes in to equalize the pressure.

Q. What does the amount of grate surface determine? A. It determines the amount of coal that can be burned per hour, and consequently, the amount of steam that can be generated.

Q. What is the object in giving a slide valve outside lap? A. To save steam by cutting off the flow of steam into the cylinder before the piston reaches the end of its stroke. For example: With 24 in. stroke of piston and $\frac{5}{8}$ cut-off, the flow of steam to the piston is cut off when the piston has moved 15 inches and it is driven the remaining 9 inches by the expansive force of the steam.

Q. What amount of refrigerating water is required for a condenser? A. For a surface condenser about 50 times, and for a jet condenser 30 times the amount of water evaporated in the boiler; more or less than these quantities being required according to the temperature of the exhaust steam.

Q. Suppose your condenser was out of order and undergoing repairs, could you run the engine? A. Yes; by attaching an exhaust pipe to the engine and exhausting into the atmosphere.

Q. With a lever safety valve, should the end of the valve stem upon which the lever rests, be square or concave? A. Neither one; it should be pointed, so that the lever will always bear directly on a line with the center of the valve stem.

Q. What is the proper proportion of a safety valve lever? A. About 7 to 1; that is, if the distance from the center of the valve to the fulcrum is 1 inch, the distance from the center of the valve to the end of the long arm of the lever should be about 7 inches.

Q. How should the grates be set in a boiler furnace? A. They should be set level, because this plan will enable the fire-

man to more easily carry a bed of fuel of uniform depth; besides, it is less laborious to clean the fire than when the grates are lower at the bridge wall.

Q. What is momentum? A. It is the product of the mass or bulk of a moving body, taken in pounds or tons, multiplied by the velocity of the moving mass, generally taken in feet per second.

Q. Will an injector work at the same steam pressure when it lifts the water as when the water flows to it under pressure? A. No; when the water flows to an injector under pressure it will work down to the lowest steam pressures, but when lifting the water it requires a steam pressure of ten pounds or over to work the injector.

Q. What is the greatest height to which an injector will lift water? A. That depends upon the starting steam pressure. There are injectors that will lift water two feet with 10 lbs. steam pressure, five feet with 30 lbs., and from 12 to 25 feet with 60 lbs. and over.

Q. If the pulley on the main shaft of an engine driving a fly-ball governor be reduced in diameter, what affect will it have on the speed of the engine? A. The speed of the engine will be increased.

Q. Which is the greater, the bursting or the collapsing pressure of a boiler tube? A. A boiler tube will collapse under less pressure than would be required to burst it.

Q. Should a horizontal externally fired boiler be set level or with a pitch? A. It is customary to set such a boiler one inch lower at the end to which the blow-off pipe is attached, in order to drain the boiler readily.

Q. In a slide valve engine with a connecting rod, will the valve cut off the same at both ends of the stroke if it has equal lap and lead? A. No; owing to the angularity of the connecting rod.

Q. Is it proper to close the damper with a banked fire? A. The damper should never be closed tightly while there is fire.

CHAPTER XXIII.

INSTRUCTIONS FOR LINING UP EXTENSION TO LINE SHAFT.

The erection of a line shaft, or an extension to one, is a job that should have the services of a competent millwright or machinist, as it is one calling for experience and considerable skill.

The following drawing will serve to illustrate the operation. A linen line or fine wire should be stretched beneath the shaft and parallel to it, and extending beyond the termination of the extension.

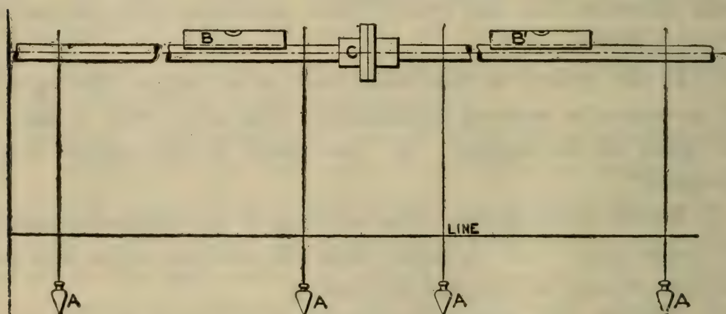


Fig. 311. Method of lining up shafting.

To set the line parallel to the main line shaft, hang the plumb-bobs *A A* over the shaft, as shown in the sketch, and then adjust the line until it just touches the lines supporting the bobs, without disturbing their position. If the plumb-bobs give trouble by swaying, set pails of water so that the bobs will be immersed; this will stop the swaying without destroying their truth. The plumb-bobs may just as well be old nuts or similar pieces of iron,

as the regulation type, since the result will be exactly the same. After getting the line adjusted to the desired position, suspend the plumb-bobs *A A* along the direction of the extension, so that their supporting cords will just touch the line without disturbing it. The new section of the shaft is now brought in position sideways until it also touches the cords of the plumb-bobs *A A* , which, of course, locates it parallel with the main shaft in a horizontal plane. To get it to the right height, enter the shaft coupling of the new part into coupling of the main shaft, and then adjust until the shaft shows level when tested with an accurate spirit level. A level suitable for this work should be of iron and planed on the under side with a V-groove, which will always locate it parallel with the shaft when testing it. Before leveling the new part of the shaft, it will be necessary to try the shaft already in position, as it may not be level. If found "out" it should be leveled, but sometimes this will not be possible or feasible, in which case it will be necessary to set the new part at the same inclination. To do this, test the main shaft and find how much it is out, and adjust the level by strips of paper until it shows "fair." The paper should be secured to the level by glue or other means and used on the new shaft in that condition, always keeping the level with the "packed" end pointing in the same direction. After getting the new part in position, it is well to test it before connecting it to the main part; that is, it should be turned by hand to determine if the frictional resistance is excessive or not. After connecting with the main part, it is not a bad idea to test it again by hand, if possible. With a long shaft it may be necessary to disconnect the farther sections and remove the belts from the connected machines. In this way a fair idea of the frictional resistance may be obtained. As before stated, this work requires experience and skill, and should properly be done by one thoroughly competent for the work; for, while his services may seem a trifle expensive, it will usually be found to pay better in the

long run, as the frictional resistance of an improperly lined shaft will quickly consume coal enough to pay the difference.

SIMPLICITY IN STEAM PIPING.

In building steam power plants, and especially in arranging the piping connections for them, simplicity is a characteristic the value of which is often too little appreciated. It should be borne in mind that extra valves and duplicate piping mean a very considerable amount of capital lying at waste to meet a contingency, which may, in all probability, never arise, not to speak of the care and attention required to keep piping and valves which are rarely used in shape for service. Another point which ought to be realized in the design of piping, is that every square foot of uncovered surface, as in flanges and the like, causes the loss of about one dollar per year in condensation of steam, and each square foot of uncovered surface represents the loss of nearly one-quarter of this amount. The principle of construction is to design the piping with the utmost simplicity possible; without any double connections, put it up so that no accidents can happen to it. It is argued that this is impossible, but it is equally impossible to insure absolute immunity against "shut downs," of greater or less duration, by any amount of duplex connections, for even the blowing out of a single gasket can blow down a whole battery of boilers before a 12-inch valve can be closed and another opened. With the more extensive introduction of high-pressure valves and fittings, it is possible, by proper design, to reduce the liability to accident very nearly to the point of absolute safety, and by the introduction of one or two extra valves, it is generally possible to divide the plant into sections, any one of which can, if occasion demands, be operated independently. No fixed rules can be laid down and the line between absolute simplicity and necessary complexity must be drawn separately for each plant with due regard to the work it has to perform, but it

should be remembered that the more simple a plant can be made to accomplish the work with absolute reliability, the greater the achievement in economy of first cost, and in availability and economy of operation.

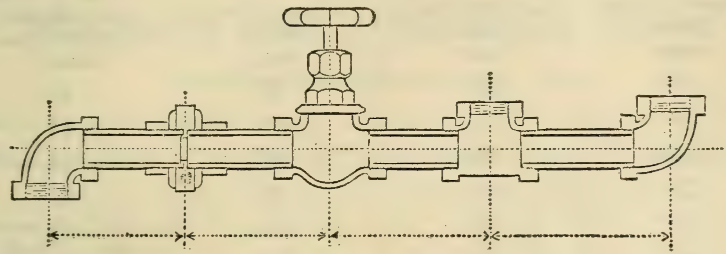


Fig. 312. Diagram showing screwed valve and fittings.

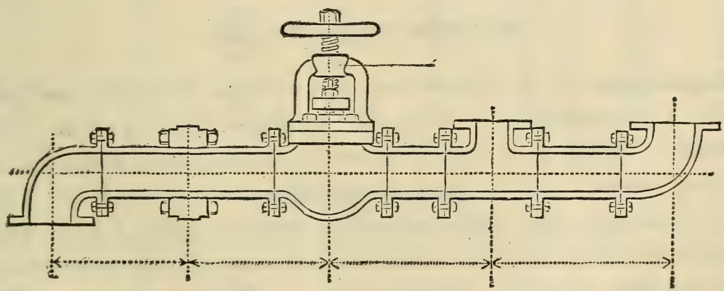


Fig. 313. Diagram showing flanged valve and fittings.

CUTTING PIPE TO ORDER.

In placing orders for pipe, a diagram should be made, according to above cuts. Great care should be taken in making a diagram for large pipe; all measurements should be from centers. When flanged fittings are used, state if desired drilled, and if with bolts and gaskets complete. If it is desired that the

fittings be made tight, then mark such pieces at point joint is desired, on diagram.

FEED-WATER REQUIRED BY SMALL ENGINES.

Pressure of Steam in Boiler, by Gauge.	Pounds of Water per Effective Horse-power per Hour.	Pressure of Steam in Boiler, by Gauge.	Pounds of Water per Effective Horse-power per Hour.
10	118	60	75
15	111	70	71
20	105	80	68
25	100	90	65
30	93	100	63
40	84	120	61
50	79	160	58

HEATING FEED-WATER.

Feed-water, as it comes from the wells or hydrants, has ordinarily a temperature of from 35° in winter to from 60 to 70° in summer. Much fuel can be saved by heating this water by the exhaust steam, whose heat would otherwise be wasted. Until quite recently, only non-condensing engines utilized feed-water heaters but lately they have been introduced with success between the cylinder and the air-pump in condensing engines. The saving in fuel due to heating feed-water is given on page 680.

RATING BOILERS BY FEED-WATER.

The rating of boilers has, since the Centennial Exposition in 1876, been generally based on 30 pounds feed-water per hour per horse-power. This is a fair average for good non-condensing engines working under about 70 to 100 ponnds pressure. But different

pressures and different rates of expansion change the requirements for feed-water. The following table gives Prof. R. H. Thurston's estimate of the steam consumption for the best classes of engines in common use when of moderate size and in good order: —

WEIGHTS OF FEED WATER AND OF STEAM.

NON-CONDENSING ENGINES. — R. H. T.

Steam Pressure.		Lbs. per H. P. per Hour. — Ratio of Expansion.					
Atmosphere.	Lbs. per sq. in.	2	3	4	5	7	10
3	45	40	39	40	40	42	45
4	60	35	34	36	36	38	40
5	75	30	28	27	26	30	32
6	90	28	27	26	25	27	29
7	105	26	25	24	23	25	27
8	120	25	24	23	22	22	21
10	150	24	23	22	21	20	20

CONDENSING ENGINES.

2	30	30	28	28	30	35	40
3	45	28	27	27	26	28	32
4	60	27	26	25	24	25	27
5	75	26	25	25	23	22	24
6	90	26	24	24	22	21	20
8	120	25	23	23	22	21	20
10	150	25	23	22	21	20	19

Small engines having greater proportional losses in friction, in leaks, in radiation, etc., and besides receiving generally less care

in construction and running than larger ones, require more feed water (or steam) per hour.

FEED WATER HEATERS.

Inattention to the temperature of feed water for boilers is entirely too common, as the saving in fuel that may be effected by thoroughly heating the feed water — by means of the exhaust steam in a properly constructed heater — would be immense, as may be seen from the following facts: A pound of feed water entering a steam boiler at a temperature of 50° Fahr., and evaporating into steam of 60 lbs. pressure, requires as much heat as would raise 1157 lbs. of water 1 degree. A pound of feed water raised from 50° Fahr. to 220° Fahr. requires 987 thermal units of heat, which if absorbed from exhaust steam passing through a heater, would be a saving of 15 per cent in fuel. Feed water at a temperature of 200° Fahr., entering a boiler, as compared in point of economy, with feed water at 50°, would effect a saving of over 13 per cent in fuel; and with a well-constructed heater there ought to be no trouble in raising the feed water to a temperature of 212° Fahr. If we take the normal temperature of the feed water at 60°, the temperature of the heated water at 212° and the boiler pressure at 20 lbs., the total heat imparted to the steam in one case is 1192.5° minus 60° = 1132.5°; and in the other case, 1192.5° minus 212° = 980.5°, the difference being 152°, or a saving of 152/1132.5 = 13.4 per cent. Supposing the feed water to enter the boiler at a temperature of 32° Fahr., each pound of water will require about 1200 units of heat to convert it into steam, so that the boiler will evaporate between $6\frac{2}{3}$ and $7\frac{1}{2}$ lbs. of water per pound of coal. The amount of heat required to convert a pound of water into steam varies with the pressure, as will be seen by the following table: —

TABLE SHOWING THE UNITS OF HEAT REQUIRED TO CONVERT ONE POUND OF WATER, AT THE TEMPERATURE OF 32° FAHR., INTO STEAM AT DIFFERENT PRESSURES.

Pressure of Steam in lbs. per Sq. In. by Gauge.	Units of Heat.	Pressure of Steam in lbs. per Sq. In. by Gauge.	Units of Heat.
1	1,148	110	1,187
10	1,155	120	1,189
20	1,161	130	1,190
30	1,165	140	1,192
40	1,169	150	1,193
50	1,173	160	1,195
60	1,176	170	1,196
70	1,178	180	1,198
80	1,181	190	1,199
90	1,183	200	1,200
100	1,185		

If the feed water has any other temperature the heat necessary to convert it into steam can easily be computed. Suppose, for instance, that its temperature is 65°, and that it is to be converted into steam having a pressure of 80 lbs. per square inch. The difference between 65 and 32 is 33; and subtracting this from 1181 (the number of units of heat required for feed water having a temperature of 32°), the remainder, 1148, is the number of units for feed water with the given temperature. Yet it must be understood that any design of heater that offers such resistance to the free escape of the exhaust steam as to neutralize the gain that would otherwise be obtained from its use, ought to be avoided, as the loss occasioned by back pressure on the exhaust, in many instances, counteracts the advantages derived from the heating of the feed water.

Feed water heaters are a most important feature of a good steam plant. First, by utilizing the heat of the exhaust steam

from the engine or waste gases in chimney, the feed water may be heated to about 210° Fahr., with ease, before entering boilers, by this means saving fuel and increasing capacity of boiler. Second. By heating the water, the boilers are protected from serious and unequal strain, as the difference of temperature between incoming water and outgoing steam may be kept about 110° (210° to 320°). Third. Every heater must necessarily be a water purifier, as the mud and lime are eliminated, to some degree at least, before the water reaches the boiler, by heat.

TABLE.

SHOWING GAIN BY USE OF FEED WATER HEATER. PERCENTAGE OF HEAT REQUIRED TO HEAT WATER FOR DIFFERENT FEED AND BOILING TEMPERATURES, AS COMPARED WITH A FEED AND BOILING TEMPERATURE OF 212°.

Boiling Point. Fahr.	Initial Temperature of feed water.										
	32°	50°	68°	86°	104°	122°	140°	158°	176°	194°	212°
212	1.19	1.17	1.15	1.13	1.11	1.10	1.08	1.06	1.04	1.02	1.00
230	1.20	1.18	1.16	1.14	1.12	1.10	1.08	1.06	1.04	1.02	1.01
248	1.20	1.18	1.16	1.14	1.13	1.11	1.09	1.07	1.05	1.03	1.01
266	1.21	1.19	1.17	1.15	1.13	1.11	1.09	1.07	1.06	1.04	1.02
284	1.21	1.20	1.18	1.16	1.14	1.12	1.10	1.08	1.06	1.04	1.02
302	1.22	1.20	1.18	1.16	1.14	1.12	1.11	1.09	1.07	1.05	1.03
320	1.22	1.21	1.19	1.17	1.15	1.13	1.11	1.09	1.07	1.05	1.03
338	1.23	1.21	1.19	1.17	1.15	1.14	1.12	1.10	1.08	1.06	1.04
356	1.23	1.22	1.20	1.18	1.16	1.14	1.12	1.10	1.08	1.06	1.04
374	1.24	1.22	1.20	1.18	1.17	1.15	1.13	1.11	1.09	1.07	1.05
392	1.24	1.23	1.21	1.19	1.17	1.15	1.13	1.11	1.09	1.07	1.06
410	1.25	1.23	1.22	1.20	1.18	1.16	1.14	1.12	1.10	1.08	1.06
428	1.25	1.24	1.22	1.20	1.18	1.16	1.14	1.12	1.11	1.09	1.07

There are two distinct types of heaters in which heat is derived from exhaust steam. These are known as closed and open heaters. Each has its advantages and disadvantages. The closed heater is constructed so that the water is forced under pres-

sure through tubes or chambers surrounded by the exhaust steam, the heat being transmitted through the walls of the tubes and chambers. The open heater is a vessel in which the feed water comes into direct contact with the exhaust steam, by spraying or intermingling. The heated water is pumped hot into the boiler. The closed heater has the advantage of permitting the water to pass through the pump cold and in that state is easily handled. To pump hot water from an open heater requires special care in piping and packing the feed pump. The closed heater, being a purifier (if any lime is present in water, a portion is bound to be precipitated by heat), should be cleaned, a job about as difficult as cleaning a boiler; or blown out, which is never a satisfactory method. In the precipitation of lime by heat, carbonic acid gas is set free and chemists say that this gas in a nascent state (just being born) attacks iron and brass. Whatever the cause, experience has demonstrated that ordinary wrought iron, steel and brass, corrode under this action. The open heater, being usually a large chamber, is accessible for cleaning out, and if made with ordinary care will last a long time. A leak in it is not a serious matter, while a leak in the closed heater means a waste of hot water into the exhaust pipe. The open heater has, at times, been the cause of serious mishaps. In it the steam and water mix; with any stoppage in exit of feed water, there is danger of flooding the cylinder of the steam engine through exhaust pipe, causing a wreck. The more modern forms of these heaters and the experience obtained in their use have reduced this difficulty to a minimum.

WATER.

Pure water at 62° F. weighs 62.355 pounds per cubic foot, or $8\frac{1}{3}$ lbs. per U. S. gallon; 7.48 gallons equal 1 cu. ft. It takes 30 lbs., or 3.6 gal. for each horse-power per hour. It would be difficult to get at the total daily horse-power of steam used in the

U. S., but it reaches into the billions of gallons of feed water per day. The importance of knowing what impurities exist in most feed waters, how these act on a boiler and how they may be removed is, therefore, patent to every intelligent engineer. We give therefore, the thoughts of some prominent investigators on the subject.

Prof. Thurston says: —

"Incrustation and sediment are deposited in boilers, the one by the precipitation of mineral or other salts previously held in solution in the feed water, the other by the deposition of mineral insoluble matters, usually earths, carried into it in suspension or mechanical admixture. Occasionably also, vegetable matter of a glutinous nature is held in solution in the feed water, and, precipitated by heat or concentration, covers the heating surfaces with a coating almost impermeable to heat, and hence, liable to cause an overheating that may be very dangerous to the structure. A powdery mineral deposit sometimes met with is equally dangerous, and for the same reason. THE ANIMAL AND VEGETABLE OILS AND GREASES CARRIED OVER FROM THE CONDENSER OR FEED WATER HEATER ARE ALSO VERY LIKELY TO CAUSE TROUBLE. Only mineral oils should be permitted to be thus introduced, and that in minimum quantity. Both the efficiency and safety of the boiler are endangered by any of these deposits.

"The amount of the foreign matter brought into the steam boiler is often enormously great. A boiler of 100 horse-power uses, as an average, probably a ton and a half of water per hour, or not far from 400 tons per month, steaming ten hours per day; and even with the water as pure as the Croton at New York, receives 90 pounds of mineral matter, and from many spring waters a *ton*, which must be either blown out or deposited. These impurities are usually either calcium carbonate or calcium sulphate, or a mixture; the first is most common on land, the second at sea. Organic matters often

harden these mineral scales and make them more difficult of removal.

"The only positive and certain remedy for incrustation and sediment, once deposited, is periodical removal by mechanical means at sufficiently frequent intervals to insure against injury by too great accumulation. Between times, some good may be done by special expedients suited to the individual case. No one process and no one antidote will suffice for all cases.

"Where carbonate of lime exists, sal-ammoniac may be used as a preventive of incrustation, a double decomposition occurring resulting in the production of ammonia carbonate and calcium chloride — both of which are soluble, and the first of which is volatile. The bicarbonate may be in part precipitated before use by heating to the boiling point, and thus breaking up the salt and precipitating the insoluble carbonate. Solutions of caustic lime and metallic zinc act in the same manner. Waters containing tannic acid and the acid juices of oak, sumach, logwood, hemlock, and other woods, are sometimes employed, but are apt to injure the iron of the boiler, as may acetic or other acid contained in the various saccharine matters often introduced into the boiler to prevent scale, and which also make the lime-sulphate scale more troublesome than when clean. Organic matter should never be used.

"The sulphate scale is sometimes attacked by the carbonate of soda, the products being a soluble sodium sulphate and a pulverulent insoluble calcium carbonate, which settles to the bottom like other sediments and is easily washed off the heating surfaces. Barium chloride acts similarly, producing barium sulphate and calcium chloride. All the alkalies are used at times to reduce incrustations of calcium sulphate, as is pure crude petroleum, the tannate of soda and other chemicals.

"The effect of incrustation and of deposits of various kinds, is to enormously reduce the conducting power of heating surfaces;

so much so, that the power, as well as the economic efficiency of a boiler, may become very greatly reduced below that for which it is rated, and the supply of steam furnished by it may become wholly inadequate to the requirements of the case.

"It is estimated that $\frac{1}{16}$ of an inch (0.16 cm.) thickness of hard scale on the heating surface of a boiler will cause a waste of nearly one-eighth of its efficiency, and the waste increases as the square of its thickness. The boilers of steam vessels are peculiarly liable to injury from this cause where using salt water, and the introduction of the surface condenser has been thus brought about as a remedy. Land boilers are subject to incrustation by the carbonate and other salts of lime and by the deposit of sand or mud mechanically suspended in the feed water.

THE TEMPERATURE AND PRESSURE OF SATURATED STEAM.

The accompanying diagram and explanation, taken from the technical publication, *The Locomotive*, will be found much more convenient for reference than steam tables. The description says that one of the most fundamental and best known facts in steam engineering is that saturated steam has a certain definite temperature for each and every definite pressure; and in all books on steam we find tables of corresponding temperatures and pressures, by the use of which we are enabled to find out what the temperature of the steam is when we know what the pressure is, and *vice versa*. For accurate work these tables are all right; but when (as is usually the case) we do not need to know either the temperature or the pressure with any very great precision, a diagram which presents the facts directly to the eye is much more convenient. Such a diagram is presented herewith. On the left-hand side of each vertical line are marked the pressures, and on the right-hand side of the same lines are marked the corresponding temperatures. The pres-

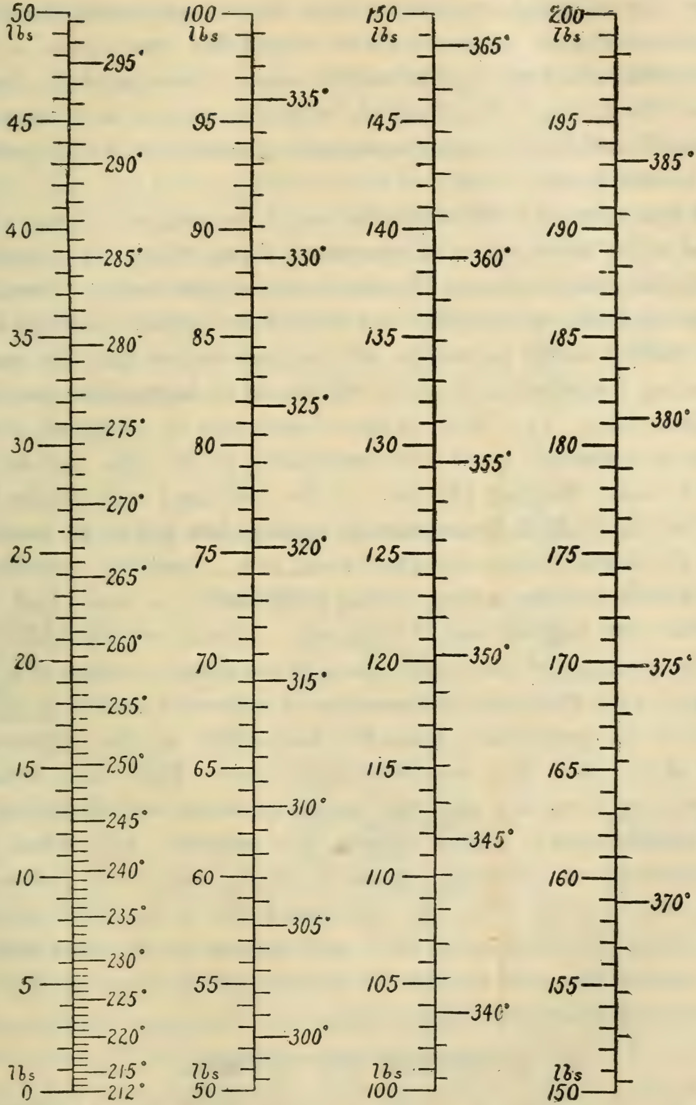


Fig. 314. Comparative diagram showing the temperature and pressure of saturated steam.

sures are all gauge pressures, that is, they represent the direct gauge reading or pressure above that of the atmosphere. The temperatures are on the Fahrenheit scale. The diagram is based upon Prof. Cecil H. Peabody's steam tables, it is therefore assumed that the average atmospheric pressure is 14.70 pounds per square inch.

A few examples will make the use of the diagram clear: (1) What is the temperature of saturated steam when its pressure, above the atmosphere, is 75 pounds per square inch? Ans. We find 75 pounds on the left-hand side of the second vertical line, and looking on the other side of the line we see that the corresponding temperature is just a fraction of a degree less than 320 degrees Fahr. (2) What is the temperature of saturated steam when its pressure, above the atmosphere, is 197 lbs. per square inch? Ans. We find 197 lbs. on the left-hand side of the last vertical line. It is not marked in figures, but 195 is so marked, and 197 is two divisions higher than 195. Looking opposite to 197 we see that the corresponding temperature is about half way between 386 degrees and 387 degrees. Hence, we conclude that the temperature of saturated steam at the given pressure is about $386\frac{1}{2}°$. (3) When the temperature of saturated steam is 227°, what is its pressure? Ans. We find 227° on the right-hand side of the first line, two divisions above 225°; and looking opposite to it, we see that the gauge pressure corresponding to this temperature is almost exactly five pounds. (4) When the temperature of saturated steam is 363°, what is its pressure? Ans. We find 363° on the right-hand side of the third vertical line, three divisions above 360°, and looking on the other side of the vertical line, we see that the corresponding gauge pressure is about $144\frac{1}{2}$ lbs. to the square inch.

SOMETHING FOR NOTHING.

In the first place, it should be remembered that in mechanics the measure of work done is the foot pound, a term which defines

itself. A foot pound of work is the amount of energy required to lift one pound one foot high. A foot pound, therefore, is the product of force and distance, force being simply a push or a pull. A machine can be made to increase the acting force, as is seen in the case of a crane, where the weight lifted is much greater than the force applied at the handle by the operator. It is also possible to increase the distance moved by some part of a machine, but it must be done by applying a greater force as in the case of a steam engine, where the distance moved by the belt is greater than the space passed over by the piston, but the total pressure of the steam against the piston is greater than the effective pull exerted by the belt.

Melting Points of Metals and Solids.

	Deg. Fahr.		Deg. Fahr.
Antimony melts at	951	Platinum melts at	4580
Bismuth "	476	Potassium "	135
Brass "	1900	Saltpeter "	600
Cadmium "	602	Steel "	2340 to 2520
Cast Iron "	1890 to 2160	Sulphur "	225
Copper "	1890	Silver "	1250
Glass "	2377	Tin "	420
Gold "	2250	Wrought Iron	2700 to 2880
Lead "	594	Zinc "	740
Ice "	32	Aluminum "	1260

In both the crane and the steam engine, however, the applied force multiplied by the distance through which it moves in a given time, must be enough greater than the product of the force at the crane hook or the rim of the fly-wheel, and the distance through

which it moves to make up for the loss through friction in the machine itself. The foot pounds of work done by any machine whatever must always be less than the foot pounds put into the machine in the same length of time. A study of this principle and of the methods of applying it, is all that is necessary to enable one to decide upon the soundness of the claims made for any power multiplying device. A British Thermal Unit (B. T. U.) is the amount of heat required to raise the temperature of a pound of water 1° Fahr., and its dynamic value is 778 lbs. raised to a height of one foot.

CHIMNEYS.

Chimneys are required for two purposes: 1st, to carry off obnoxious gases; 2d, to produce a draft, and so facilitate combustion. The first requires size, the second, height. Each pound of coal burned yields from 13 to 30 pounds of gas, the volume of which varies with the temperature. The weight of gas to be carried off by a chimney, in a given time, depends on three things — size of chimney, velocity of flow and density of gas. But as the density decreases directly as the absolute temperature, while the velocity increases with a given height, nearly as the square root of the temperature, it follows that there is a temperature at which the weight of gas delivered is a maximum. This is about 550° above the surrounding air. Temperature, however, makes so little difference that at 550° above the quantity is only 4 per cent greater than at 300°. Therefore, height and area are the only elements necessary to consider in an ordinary chimney. The intensity of draft is, however, independent of the size, and depends upon the difference in weight of the outside and inside columns of air, which varies nearly as the product of the height into the difference of temperature. This is usually stated in an equivalent column of water, and may vary from 0 to possibly 2 inches. After a height has been reached to produce draft of sufficient

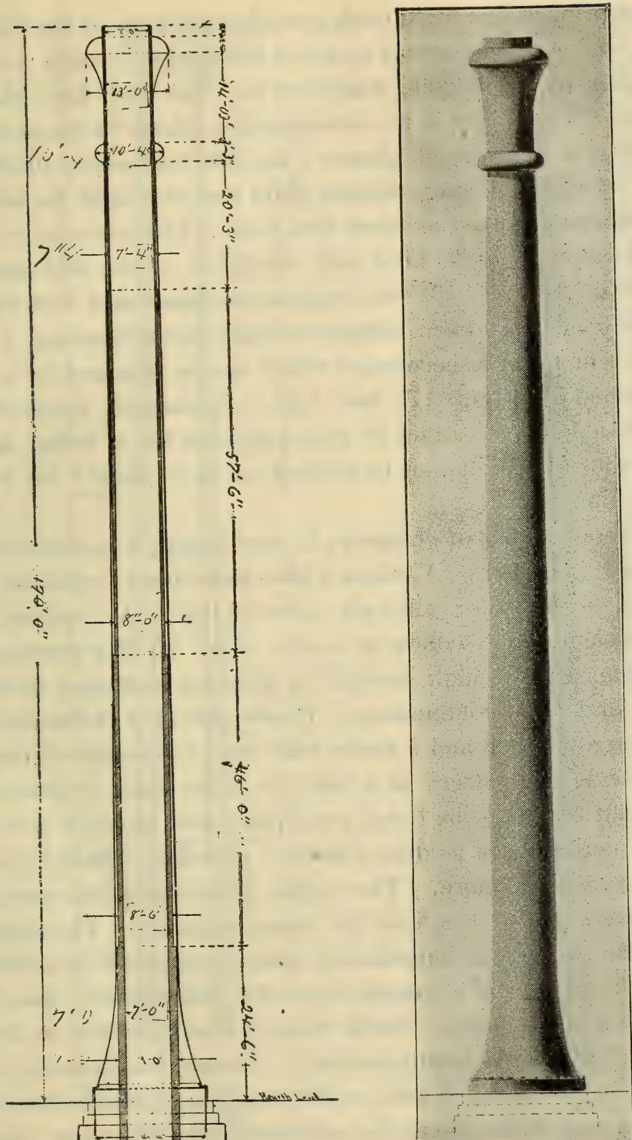


Fig. 315. Section and elevation of steel stack.

intensity to burn fine, hard coal, provided the area of the chimney is large enough, there seems no good mechanical reason for adding further to the height, whatever the size of the chimney required. Where cost is no consideration, there is no objection to building as high as one pleases; but for the purely utilitarian purpose of steam making, equally good results might be attained with a shorter chimney at much less cost. The intensity of draft required varies with the kind and condition of the fuel and the thickness of the fires. Wood requires the least, and fine coal or slack the most. To burn anthracite slack to advantage, a draft of $1\frac{1}{4}$ inch of water is necessary, which can be attained by a well-proportioned chimney 175 feet high. Generally, a much less height than 100 feet cannot be recommended for a boiler, as the lower grades of fuel cannot be burned as they should be with a shorter chimney.

The proportioning of chimneys is very largely a matter of experience and judgment. Various rules have been formulated for this purpose, but they all vary more or less. A chimney must have sufficient cross-section to easily carry off the products of combustion, and be high enough to produce sufficient draft for complete and rapid combustion. Where there is a choice between a high narrow stack and a lower wide one, the nature of the fuel should decide the matter; as a rule, the taller stack is preferable. The amount of fuel to be burnt per square foot of grate per hour has been increasing in modern practice; therefore, the old rules do not fit the case any more. Then again, it makes a difference how many boilers are to run into the same chimney. The heaviest work of the chimney is immediately after firing, since the friction through the fresh coal is greater and the temperature less then than some minutes later. But it would be bad practice to fire all boilers or all doors simultaneously. Hence, the second boiler does not require as much area as the first; say, 75 per cent will do. After that there comes the additional consideration that as

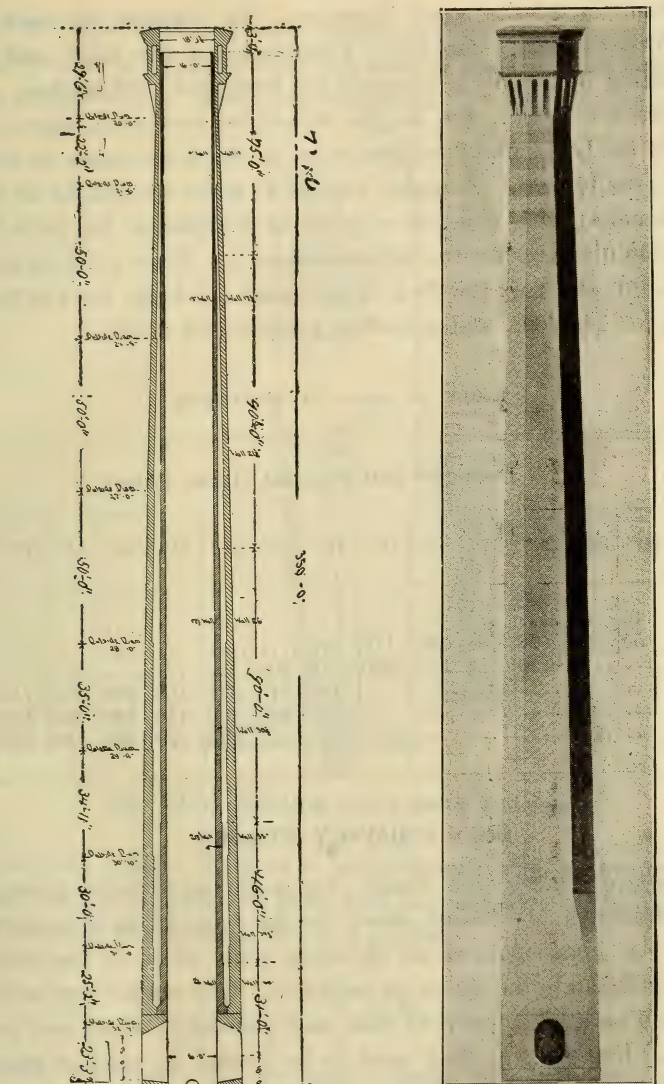


Fig. 316. Section and elevation of brick stack.

the diameter of the stack increases, the friction in stack and breeching decreases rapidly. Therefore, for the third and each succeeding boiler, 50 per cent of the first area will suffice. But as more are added, the height should be increased, more especially if the horizontal flue from boiler to stack increases in length, as it usually will. A good rule is to make the height 25 times the diameter, with possibly a gradual decrease in the ratio to 20 times the diameter for the larger chimneys. Thus a 4-foot diameter would call for 100 feet height, and a 5-foot, for 120 feet, a 6-foot for 140 feet, and a 10-foot for 200 feet height.

TABLE OF SIZES OF CHIMNEYS.

Height.	Diameter and Nominal Horse Power.													
	20″	26″	30″	34″	36″	40″	44″	50″	54″	58″	60″	64″	72″	78″
70 ft.	40	60	....	....	....	....	....	....	....	....	....	....	....	....
80 ft.	50	75	100	130	150	175	200	....	....	....	....	....	....	....
90 ft.	....	....	120	150	175	200	225	300	....	....	....	....	....	....
100 ft.	....	....	....	....	....	....	250	340	375	430	500	600	750	930
110 ft.	....	....	....	....	....	....	....	360	400	455	550	650	825	990
120 ft.	....	....	....	....	....	....	....	....	425	500	600	700	900	1050

IRON CHIMNEY STACKS.

In many places iron stacks are preferred to brick chimneys. Iron chimneys are bolted down to the base so as to require no stays. A good method of securing such bolts to the stack is shown in detail in the figure on page 693. Iron stacks require to be kept well painted to prevent rust, and generally, where not bolted down, as here shown, they need to be braced by rods or wires to surrounding objects. With four such braces attached to an angle iron ring at $\frac{2}{3}$ the height of stack, and spreading laterally at

least an equal distance, each brace should have an area in square inches equal to $\frac{1}{1000}$ the exposed area of stack (dia. x height) in feet. Stability or power to withstand the overturning force of

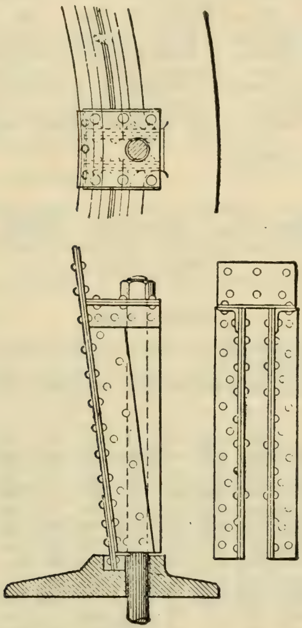


Fig. 317. Holding down bolts and lugs.

the highest winds, requires a proportionate relation between the weight, height, breadth of base, and exposed area of the chimney. This relation is expressed in the equation

$$C\frac{dh^2}{b} = W,$$

in which d equals the average breadth of the shaft; $h =$ its height; $b =$ the breadth of base — all in feet; $W =$ weight of chimney in lbs., and $C =$ a coefficient of wind pressure per

square foot of area. This varies with the cross-section of the chimney, and = 56 for a square, 35 for an octagon and 28 for a round chimney. Thus a square chimney of average breadth of 8 feet, 10 feet wide at base and 100 feet high, would require to weigh 56 x 8 x 100 x 10 = 448,000 lbs., to withstand any gale likely to be experienced. Brickwork weighs from 100 to 130 lbs. per cubic foot; hence, such a chimney must average 13 inches thick to be safe. A round stack could weigh half as much, or have less base.

WEIGHT OF SHEET LAP RIVETED STEEL SMOKE STACKS, PER FOOT.

THICKNESS.

DIA.	No. 18	No. 16	No. 14	No. 12	No. 10	No. 8	3/16″	7/32″	1/4″	9/32″	5/16″	11/32″	3/8″	13/32″	7/16″	15/32″	1/2″
12″	8	10	13	17	21	25½	31¼	37	42	47	52½	58	63	68¼	73½	78¾	84
14″	9¼	11½	15¼	20	24½	29¾	36¾	42	48½	54½	62½	67	73½	79¼	85	91	97
16″	10¼	13	17½	23	28	34	42	49	56	63	70	77	84	91	98	105	112
18″	11¾	14½	19¾	26	31½	38¼	47	55	63	71	79	86	94	102	110	118	126
20″	13	16	22	28¾	35	42½	52	60	69	78	86	95	104	113	121	131	138
22″	14¼	17¾	24¼	31½	38½	46¾	54	63½	73	82	91	99	108	118	127	137	146
24″	15½	19½	26½	34¼	42	51	59	68½	78½	88	98	108	118	128	137	147	157
26″	16¾	21	28¾	37	45½	55¼	63	73½	84	94	105	115	126	137	147	158	168
28″	18	22¾	31	40	49	59½	67	78	89½	100	111	122	134	145	156	167	179
30″	....	24½	33	42¾	52½	63¾	71	83	95	106½	118	130	142	154	166	178	190
32″	....	26¼	35	45½	56	68	75	87½	100½	113	125	138	150	163	175	188	201
34″	..	28	37	48¼	59½	72¼	80	93	106	119	132	146	160	173	186	199	212
36″	....	29¾	39	51	63	76½	85	100	114	128	143	158	173	188	202	216	230
38″	....	31½	41¼	53¾	66½	80¾	90	105	120	135	151	166	182	198	213	227	242
40″	....	33¼	43½	56½	70	85	94	110	126	142	158	174	191	208	224	239	254
42″	....	35	45¾	59½	73½	89¼	98	115	132	149	166	183	200	217	234	250	266
44″	....	36¾	48	62	77	93½	103	121	138	155	173	191	209	227	245	262	279
46″	....	38½	50¼	65	80½	97¾	107	126	144	162	181	199	218	237	255	273	291
48″	....	40	52½	68	84	102	112	131	150	169	188	208	227	247	266	284	303
50″	....	...	54¾	71	87½	106¼	116	136	156	176	195	216	236	258	277	296	315
52″	....	....	57	74	91	110½	121	142	162	182	203	224	245	266	287	307	328
54″	....	....	..	77	94½	114¾	124	147	168	189	211	233	254	276	298	319	349
56″	..	....	....	80	98	119	133	158	180	202	225	248	270	294	317	340	363
58″	....	....	..	83	102	123¼	137	164	186	209	232	256	280	304	327	351	375
60″	....	....	....	86	106	127½	142	169	192	215	240	264	289	314	338	362	387
62″	....	....	....	89	110	131¾	146	174	198	222	247	273	298	324	349	374	400
64″	....	....	....	92	114	136	151	179	204	229	255	281	307	333	359	385	412

CHAPTER XXIV.

HORSE-POWER OF GEARS.

To determine the horse-power, which any gear-wheel will transmit, four facts are required to be known: —

1st. The kind of wheel, whether spur, bevel, spur mortise, or bevel mortise. 2d. The pitch. 3d. The face. 4th. The velocity of pitch circle in feet per second.

Generally, the fourth fact is not known. It can be found if the pitch diameter of the wheel in inches and the number of revolutions per minute are given, for it can be obtained from them by the following rule: —

Rule. — Given the pitch diameter in inches and the number of revolutions per minute; to find the velocity of pitch line in feet per second.

First, multiply the pitch diameter (in inches) by the number of revolutions per minute. Second, divide the product thus found by 230. The quotient is the velocity required.

Example. — What is the velocity of the pitch circle of a gear-wheel in feet per second, the pitch diameter = 43 inches, the revolutions per minute = 125?

43 x 125 divided by 230 = 23.4 feet per second.

Table A shows the greatest horse-power, which different kinds of gears of 1-inch pitch and 1-inch face will safely transmit at various pitch-line velocities. To find the greatest horse-power which any other pitch and face will safely transmit, the following rule can be used: —

Rule. — Given, the pitch (in inches), face (in inches), velocity of pitch circle (in feet per second), and kind of gear; to find the greatest horse-power that can be safely transmitted.

First. Find the horse-power in Table B, which the given kind

of wheel with 1-inch pitch and 1-inch face will transmit at the given velocity. Second. Multiply the pitch by the face. Third. Multiply the horse-power found by the product of pitch by face. The final product is the horse-power required.

Example. — What is the greatest horse-power that a bevel-wheel, 43″ pitch diameter, 2″ pitch, 6″ face, and 125 revolutions per minute will safely transmit?

From previous example, we have found the pitch-line velocity to be 23.4 feet per second, which is nearest to a velocity of 24 feet per second in Table A.

First, the horse-power which a bevel wheel of 1″ pitch and 1′ face will transmit is (from table) at this velocity 4.931.

Second, the product of pitch by face is 2 x 6 = 12.

Third, 12 x 4.931 = 59.17 horse-power. Answer.

Whenever it is desirable to know about the average horse-power that any wheel will transmit, $\frac{2}{3}$ or $\frac{1}{2}$ of the results obtained by the rule above should be taken.

TABLE A. — TABLE SHOWING THE HORSE-POWER WHICH DIFFERENT KINDS OF GEAR WHEELS OF ONE INCH PITCH AND ONE INCH FACE WILL TRANSMIT AT VARIOUS VELOCITIES OF PITCH CIRCLE.

1	2	3	4	5
Velocity of pitch circle in ft. per sec.	Spur Wheels.	Spur Mortise Wheels.	Bevel Wheels.	Bevel Mortise Wheels.
2	1.338	.647	.938	.647
3	1.756	.971	1.227	.856
6	2.782	1.76	1.76	1.363
12	4.43	3.1	3.1	2.16
18	5.793	4.058	4.058	2.847
24	7.052	4.931	4.931	3.447
30	8.182	5.727	5.727	4.036
36	9.163	6.314	6.414	4.516
42	10.156	7.102	7.102	4.963
48	10.683	7.680	7.680	5.411

NOTE. — When velocities are given, which are between those in table, the horse-power can be found by interpolation.

Thus, the horse-power for spur wheels at 14 feet velocity is found as follows: —

$$\left.\begin{array}{l}14 \text{ minus } 12 = 2 \\ 18 \quad \text{“} \quad 12 = 6\end{array}\right\} 5.793 \text{ minus } 4.43 = 1.363.$$

Then $\frac{2}{6}$ of $1.363 = .454$ and $.454 + 4.43 = 4.884$ horse-power.

TABLE B. — SHAFTING. — HORSE-POWER TRANSMITTED BY VARIOUS SHAFTS, AT 100 REVOLUTIONS PER MINUTE UNDER VARIOUS CONDITIONS.

1 Diameter of Shaft.	2 Line Shafts.	3 Shaft as a Prime Mover.	4 Shafts Under Slight Bending Strain.	1 Diameter of Shaft.	2 Line Shafts.	3 Shaft as a Prime Mover.	4 Shafts Under Slight Bending Strain.
$\frac{15}{16}''$	.7	.4	1.3	$3\frac{11}{16}''$	40.	20.	80.
$1\frac{3}{16}''$	1.3	.7	2.6	$3\frac{15}{16}''$	49.	25.	97.
$1\frac{7}{16}''$	2.4	1.2	4.7	$4\frac{7}{16}''$	70.	35.	139.
$1\frac{11}{16}''$	3.8	1.9	7.6	$4\frac{15}{16}''$	96.	48.	192.
$1\frac{15}{16}''$	5.8	2.9	11.5	$5\frac{7}{16}''$	126.	64.	256.
$2\frac{3}{16}''$	8.3	4.2	16.6	$5\frac{15}{16}''$	167.	84.	334.
$2\frac{7}{16}''$	11.5	5.8	23.	$6\frac{15}{16}''$	266.	133.	532.
$2\frac{11}{16}''$	15.5	7.8	31.	$7\frac{15}{16}''$	399.	200.	797.
$2\frac{15}{16}''$	20.	10.	40.	$8\frac{15}{16}''$	570.	285.	1139.
$3\frac{3}{16}''$	26.	13.	51.	$9\frac{15}{16}''$	783.	392.	1566.
$3\frac{7}{16}''$	33.	17.	65.				

This table gives the horse-power that various sizes of shafts will safely transmit at 100 revolutions per minute under various conditions.

Prime movers are those shafts in which the variation above and below the average horse-power transmitted is great, also where the transverse strain due to belts or heavy pulleys is large, such as jack-shafts, crank-shafts, etc.

WHEEL GEARING.

The pitch line of a wheel is the circle upon which the pitch is measured, and it is the circumference by which the diameter, or the velocity of the wheel, is measured. The pitch is the arc of the circle of the pitch line, and is determined by the number of teeth in the wheel. The true pitch (chordal), or that by which the dimensions of the tooth of a wheel are alone determined, is a straight line drawn from the centers of two contiguous teeth upon the pitch line. The line of centers is the line between the centers of two wheels. The radius of a wheel is the semi-diameter running to the periphery of a tooth. The pitch radius is the semi-diameter running to the pitch line. The length of a tooth is the distance from its base to its extremity. The breadth of a tooth is the length of the face of wheel. The teeth of wheels should be as small and numerous as is consistent with strength. When a pinion is driven by a wheel, the number of teeth in the pinion should not be less than eight. When a wheel is driven by a pinion, the number of teeth in the pinion should not be less than ten. The number of teeth in a wheel should always be prime to the number of the pinion; that is, the number of teeth in the wheel should not be divisible by the number of teeth in the pinion, without a remainder. This is in order to prevent the same teeth coming together so often as to cause an irregular wear of their faces. An odd tooth introduced into a wheel is termed a hunting-tooth or cog.

TO COMPUTE THE PITCH OF A WHEEL.

Rule. — Divide the circumference at the pitch-line by the number of teeth.

Example. — A wheel 40 in. in diameter, requires 75 teeth; what is its pitch?

$$\frac{3.1416 \times 40}{75} = 1.6755 \text{ in.}$$

TO COMPUTE THE CHORDAL PITCH.

Rule. — Divide 180° by the number of teeth, ascertain the sin. of the quotient, and multiply it by the diameter of the wheel.

Example. — The number of teeth is 75 and the diameter 40 in.; what is the true pitch?

$\frac{180}{75} = 2^\circ\,24'$ and sin. of $2^\circ\,24' = .04188$, which $\times\,40 = 1.6752$ in.

TO COMPUTE THE DIAMETER OF A WHEEL.

Rule. — Multiply the number of teeth by the pitch, and divide the product by 3.1416.

Example. — The number of teeth in a wheel is 75, and the pitch 1.675 in.; what is the diameter of it?

$$\frac{75 \times 1.675}{3.1416} = 40 \text{ in.}$$

TO COMPUTE THE NUMBER OF TEETH IN A WHEEL.

Rule. — Divide the circumference by the pitch.

TO COMPUTE THE DIAMETER WHEN THE TRUE PITCH IS GIVEN.

Rule. — Multiply the number of teeth in the wheel by the true pitch, and again by .3184.

Example. — Take the elements of the preceding case.

$$75 \times 1.6752 \times .3184 = 40 \text{ in.}$$

TO COMPUTE THE NUMBER OF TEETH IN A PINION OR FOLLOWER TO HAVE A GIVEN VELOCITY.

Rule. — Multiply the velocity of the driver by its number of teeth, and divide the product by the velocity of the driven.

Example. — The velocity of a driver is 16 revolutions, the number of its teeth 54, and the velocity of the pinion is 48; what is the number of its teeth?

$$\frac{16 \times 54}{48} = 18 \text{ teeth.}$$

2. A wheel having 75 teeth is making 16 revolutions per minute. What is the number of teeth required in the pinion to make 24 revolutions in the same time?

$$\frac{16 \times 75}{24} = 50 \text{ teeth.}$$

TO COMPUTE THE PROPORTIONAL RADIUS OF A WHEEL OR PINION.

Rule. — Multiply the length of the line of centers by the number of teeth in the wheel for the wheel, and in the pinion for the pinion, and divide by the number of teeth in both the wheel and the pinion.

TO COMPUTE THE DIAMETER OF A PINION, WHEN THE DIAMETER OF THE WHEEL AND NUMBER OF TEETH IN THE WHEEL AND PINION ARE GIVEN.

Rule. — Multiply the diameter of the wheel by the number of teeth in the pinion, and divide the product by the number of teeth in the wheel.

Example. — The diameter of a wheel is 25 in., the number of its teeth 210, and the number of teeth in the pinion 30; what is the diameter of the pinion?

$$\frac{25 \times 30}{210} = 3.57 \text{ in.}$$

TO COMPUTE THE CIRCUMFERENCE OF A WHEEL.

Rule. — Multiply the number of teeth by their pitch.

TO COMPUTE THE REVOLUTIONS OF A WHEEL OR PINION.

Rule. — Multiply the diameter or circumference of the wheel or the number of its teeth, as the case may be, by the number of its revolutions, and divide the product by the diameter, circumference, or number of teeth in the pinion.

Example. — A pinion 10 in. in diameter is driven by a wheel

2 ft. in diameter, making 46 revolutions per minute; what is the number of revolutions of the pinion?

$$\frac{2 \times 12 \times 46}{10} = 110.4 \text{ revolutions.}$$

TO COMPUTE THE RELATIVE VELOCITY OF A PINION.

Rule. — Divide the diameter, circumference or number of teeth in the driver, as the case may be, by the diameter, etc., of the pinion.

WHEN THERE IS A SERIES OR TRAIN OF WHEELS AND PINIONS.

Rule. — Divide the continued product of the diameter, circumference, or number of teeth in the wheels by the continued product of the diameter, etc., of the pinions.

Example. — If a wheel of 32 teeth drive a pinion of 10, upon the axis of which there is one of 30 teeth, driving a pinion of 8, what are the revolutions of the last?

$$\frac{32}{10} \times \frac{30}{8} = \frac{960}{80} = 12 \text{ revolutions.}$$

Ex. 2. — The diameters of a train of wheels are 6, 9, 9, 10 and 12 in.; of the pinions, 6, 6, 6, 6, and 6 in.; and the number of revolutions of the driving shaft or prime mover is 10; what are the revolutions of the last pinion?

$$\frac{6 \times 9 \times 9 \times 10 \times 12 \times 10}{6 \times 6 \times 6 \times 6 \times 6} = \frac{583200}{7776} = 75 \text{ revolutions.}$$

TO COMPUTE THE PROPORTION THAT THE VELOCITIES OF THE WHEELS IN A TRAIN WOULD BEAR TO ONE ANOTHER.

Rule. — Subtract the less velocity from the greater, and divide the remainder by one less than the number of wheels in the train; the quotient is the number, rising in arithmetical progression from the less to the greater velocity.

Example. — What should be the velocities of three wheels to produce 18 revolutions, the driver making 3?

$$\frac{18 \text{ minus } 3 = 15}{3 \text{ minus } 1 = 2} = 7.5 = \text{number to be added to velocity of the}$$

driver $= 7.5 + 3 = 10.5$ and $10.5 + 7.5 = 18$ revolutions. Hence, 3, 10.5 and 18 are the velocities of the three wheels.

GENERAL ILLUSTRATIONS.

1. A wheel 96 inches in diameter, making 42 revolutions per minute, is to drive a shaft 75 revolutions per minute, what should be the diameter of the pinion?

$$\frac{96 \times 42}{75} = 53.76 \text{ in.}$$

2. If a pinion is to make 20 revolutions per minute, required the diameter of another to make 58 revolutions in the same time. 58 divided by $20 = 2.9 =$ the ratio of their diameters. Hence if one to make 20 revolutions is given a diameter of 30 in., the other will be 30 divided by $2.9 = 10.345$ in.

3. Required the diameter of a pinion to make $12\frac{1}{2}$ revolutions in the same time as one of 32 in. diameter making 26.

$$\frac{32 \times 26}{12.5} \quad 66.56 \text{ in.}$$

4. A shaft making 22 revolutions per minute, is to drive another shaft at the rate of 15, the distance between the two shafts upon the line of centers is 45 in.; what should be the diameter of the wheels?

Then, 1st, $22 + 15 : 22 : : 45 : 26.75 =$ inches in the radius of the pinion.

2d. $22 + 15 : 15 : : 45 : 18.24 =$ inches in the radius of the spur.

5. A driving shaft, making 16 revolutions per minute, is to drive a shaft 81 revolutions per minute, the motion to be communicated by two geared wheels and two pulleys, with an intermediate shaft; the driving wheel is to contain 54 teeth, and the

driving pulley upon the driven shaft is to be 25 in. in diameter; required the number of teeth in the driven wheel, and the diameter of the driven pulley. Let the driven wheel have a velocity of $\sqrt{16 \times 81} = 36$ a mean proportional between the extreme velocities 16 and 81.

Then, 1st, 36 : 16 : : 54 : 24 = teeth in the driven wheel.

2d. 81 : 36 : : 25 : 11.11 = inches diameter of the driven pulley.

6. If, as in the preceding case, the whole number of revolutions of the driving shaft, the number of teeth in its wheel and the diameter of the pulley are given, what are the revolutions of the shafts?

Then, 1st, 18 : 16 : : 54 : 48 = revolutions of the intermediate shaft.

2d. 15 : 48 : : 25 : 80 = revolutions of the driven shaft.

TO COMPUTE THE DIAMETER OF A WHEEL FOR A GIVEN PITCH AND NUMBER OF TEETH.

Rule. — Multiply the diameter in the following table for the number of teeth by the pitch, and the product will give the diameter at the pitch circle.

Example. — What is the diameter of a wheel to contain 48 teeth of 2.5 in. pitch?

$$15.29 \times 2.5 = 38.225 \text{ in.}$$

TO COMPUTE THE PITCH OF A WHEEL FOR A GIVEN DIAMETER AND NUMBER OF TEETH.

Rule. — Divide the diameter of the wheel by the diameter in the table for the number of teeth, and the quotient will give the pitch.

Example. — What is the pitch of a wheel when the diameter of it is 50.94 in., and the number of its teeth 80?

$$\frac{50.94}{25.47} = 2 \text{ in.}$$

PITCH OF WHEELS.

A TABLE WHEREBY TO COMPUTE THE DIAMETER OF A WHEEL FOR A GIVEN PITCH, OR THE PITCH FOR A GIVEN DIAMETER.

From 8 to 192 teeth.

No. of Teeth.	Diameter.	No. of Teeth.	Diameter.	No. of Teeth.	Diameter.	No. of Teeth.	Diameter.	No. of Teeth	Diameter.
8	2.61	45	14.33	82	26.11	119	37.88	156	49.66
9	2.93	46	14.65	83	26.43	120	38.2	157	49.98
10	3.24	47	14.97	84	26.74	121	38.52	158	50.3
11	3.55	48	15.29	85	27.06	122	38.84	159	50.61
12	3.86	49	15.61	86	27.38	123	39.16	160	50.93
13	4.18	50	15.93	87	27.7	124	39.47	161	51.25
14	4.49	51	16.24	88	28.02	125	39.79	162	51.57
15	4.81	52	16.56	89	28.33	126	40.11	163	51.89
16	5.12	53	16.88	90	28.65	127	40.43	164	52.21
17	5.44	54	17.2	91	28.97	128	40.75	165	52.52
18	5.76	55	17.52	92	29.29	129	41.07	166	52.84
19	6.07	56	17.8	93	29.61	130	41.38	167	53.16
20	6.39	57	18.15	94	29.93	131	41.7	168	53.48
21	6.71	58	18.47	95	30.24	132	42.02	169	53.8
22	7.03	59	18.79	96	30.56	133	42.34	170	54.12
23	7.34	60	19.11	97	30.88	134	42.66	171	54.43
24	7.66	61	19.42	98	31.2	135	42.98	172	54.75
25	7.98	62	19.74	99	31.52	136	43.29	173	55.07
26	8.3	63	20.06	100	31.84	137	43.61	174	55.39
27	8.61	64	20.38	101	32.15	138	43.93	175	55.71
28	8.93	65	20.7	102	32.47	139	44.25	176	56.02
29	9.25	66	21.02	103	32.79	140	44.57	177	56.34
30	9.57	67	21.33	104	33.11	141	44.88	178	56.66
31	9.88	68	21.65	105	33.43	142	45.2	179	56.98
32	10.2	69	21.97	106	33.74	143	45.52	180	57.23
33	10.52	70	22.29	107	34.06	144	45.84	181	57.62
34	10.84	71	22.61	108	34.38	145	46.16	182	57.93
35	11.16	72	22.92	109	34.7	146	46.48	183	58.25
36	11.47	73	23.24	110	35.02	147	46.79	184	58.57
37	11.79	74	23.56	111	35.34	148	47.11	185	58.89
38	12.11	75	23.88	112	35.65	149	47.43	186	59.21
39	12.43	76	24.2	113	35.97	150	47.75	187	59.53
40	12.74	77	24.52	114	36.29	151	48.07	188	59.84
41	13.06	78	24.83	115	36.61	152	48.39	189	60.16
42	13.38	79	25.15	116	36.93	153	48.7	190	60.48
43	13.7	80	25.47	117	37.25	154	49.02	191	60.81
44	14.02	81	25.79	118	37.56	155	49.34	192	61.13

TO COMPUTE THE STRESS THAT MAY BE BORNE BY A TOOTH.

Rule. — Multiply the value of the material of the tooth to resist transverse strain, as estimated for this character of stress, by the breadth and square of its depth, and divide the product by the extreme length of it in the decimal of a foot.

TO COMPUTE THE NUMBER OF TEETH OF A WHEEL FOR A GIVEN DIAMETER AND PITCH.

Rule. — Divide the diameter by the pitch, and opposite to the quotient in the preceding table is given the number of teeth.

TEETH OF WHEELS.

Epicycloidal. — In order that the teeth of the wheels and pinions should work evenly and without unnecessary rubbing friction, the face (from pitch line to top) of the outline should be determined by an epicycloidal curve, and the flank (from pitch line to base) by an hypocycloidal. When the generating circle is equal to half the diameter of the pitch circle, the hypocycloid described by it is a straight diametrical line, and consequently the outline of a flank is a right line and radial to the center of the wheel. If a like generating circle is used to describe face of a tooth of other wheel or pinion respectively, the wheel and pinion will operate evenly.

Involute. — Teeth of two wheels will work truly together when surfaces of their face is an involute; and that two such wheels should work truly, the circles from which the involute lines for each wheel are generated must be concentric with the wheels, with diameters in the same ratio as those of the wheels.

Curves of teeth. — In the pattern shop, the curves of epicycloidal or involute teeth are defined by rolling a template of the generating circle on a template corresponding to the pitch line, a scriber on the periphery of the template being used to define

the curve. Least number of teeth that can be employed in pinions having teeth of following classes, are: involute, 25; epicycloidal, 12; staves or pins, 6.

CONSTRUCTION OF GEARING.

If the dimensions of two wheels are determined, as well as the size of the teeth and spaces, the wheel is drawn as shown in figure. The starting-point for the division of the wheels is where the two pitch circles meet in *A*. It is advisable to determine the exact diameters of the wheels by calculation, if the difference between them is remarkable; for any division upon two circles of unequal size by means of a divider,

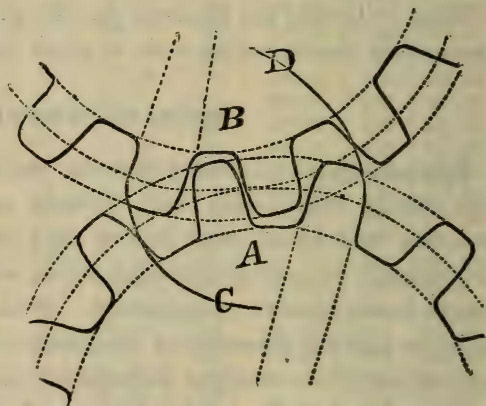


Fig. 318. Involute gear teeth.

is incorrect, because the latter measures the chord instead of the arc. From the point *A* we construct the epicycloid *C*, by rolling the circle *A* upon *B*, as its base line. That short piece of the epicycloid, from the pitch line to the face of the tooth, is the curvature for that part of the tooth and the wheel *B*. This curvature obtained for one side of the tooth, serves for both sides of it, and also for all the teeth in the wheel. The lower part of the tooth, or that inside the pitch-line, is immaterial to the working of the wheel; this may be a straight line, as shown by the dotted lines which are in the direction of the diameters, or may be a curved line, as is seen in the wheel *A*. This line must be so formed as not to touch the upper or curved part of the tooth. The root of

the tooth, or that part of it which is connected with the rim of the wheel, is the weakest part of the tooth, and may be strengthened by filling the angles at the corners. The curvature for the teeth in the wheel A is found in a similar manner to that of B. The pitch circle A serves now as a base line, and the circle B is rolled upon it, to obtain the circle D. This line forms the curvature for the teeth of A, and serves for all the teeth in A — also for both sides of the teeth. In most practical cases the curvature of the teeth is described as a part of a circle, drawn from the center of the next tooth, or from a point more or less above or below that center, or the radius greater or less in strength than the pitch of the wheel. Such circles are never correct curves, and no rule can be established by which their size and center meets the form of the epicycloid.

BEVEL WHEELS.

If the lines C A and B C represent the prolonged axes, which are to revolve with different or similar velocities, the position and sizes of the wheels for driving these axes are determined by the distance of the wheels from the point C. The diameters of the wheels are as the angles a and b and inversely as the number of revolutions. These angles are, therefore, to be determined before the wheels can be drawn.

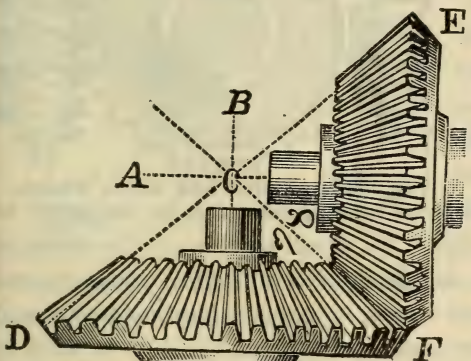


Fig. 319. Bevel gears.

By measuring the distances from C to the line E, or from C to F, the sizes of the wheels are determined. These lines E, F and D F, are the diameters for the pitch lines; from them the form

of the tooth is described on the beveled face of the wheel. If the form of the tooth is described on the largest circle of the wheel, all the lines from this face run to the point C, so that when the wheel revolves around its axis, all the lines from the teeth concentrate in the point C, and form a perfect cone. Curvature, thickness, length and spaces are here calculated as on face wheels; the thickness is measured in the middle of the width of the wheel.

WORM-SCREW.

If a single screw A works in a toothed wheel, each revolution of the screw will turn the wheel one cog; if the screw is formed of more than one thread, a corresponding number of teeth will be moved by each revolution. With the increase of the number of threads, the side motion of the wheel and screw is accelerated; and when the threads and number of teeth are equal, an angle of 45° is required for teeth and thread, provided their diameters also are equal. This motion causes a great deal of friction and it is only resorted to where no other means can be employed to produce the required motion. In small machinery, the worm is frequently made use of to produce a uniform, uninterrupted motion; the screw, in such cases, is made of hardened steel and the teeth of the wheel are cut by the screw which is to work in the wheel. If the form of the teeth in the wheel is not curved and its face is concave so as to fit the thread in all points, the screw will touch the teeth but in one point and cause them to be liable to breakage.

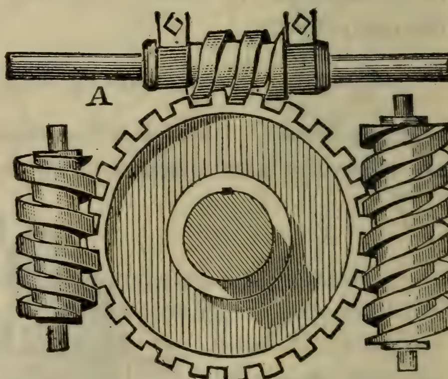


Fig. 320. Worm and worm wheel.

PROPORTIONS OF TEETH OF WHEELS.

Tooth. — In computing the dimensions of a tooth, it is to be considered as a beam fixed at one end, the weight suspended from the other, or face of the beam; and it is essential to consider the element of velocity, as its stress in operation, at high velocity with irregular action, is increased thereby. The dimensions of a tooth should be much greater than is necessary to resist the direct stress upon it, as but one tooth is proportioned to bear the whole stress upon the wheel, although two or more are actually in contact at all times; but this requirement is in consequence of the great wear to which a tooth is subjected, the shocks it is liable to from lost motion when so worn as to reduce its depth and uniformity of bearing, and the risk of the breaking of a tooth from a defect. A tooth running at a low velocity may be materially reduced in its dimensions compared with one running at high velocity and with a like stress. The result of operations with toothed wheels, for a long period of time, has determined that a tooth with a pitch of 3 inches and a breadth 7.5 inches will transmit, at a velocity of 6.66 feet per second, the power of 59.16 horses.

TO COMPUTE THE DEPTH OF A CAST-IRON TOOTH.

1. When the stress is given.

Rule. — Extract the square root of the stress, and multiply it by .02.

Example. — The stress to be borne by a tooth is 4886 lbs.; what should be its depth?

$$\sqrt{4886} \times .02 = 1.4 \text{ in.}$$

2. When the horse-power is given.

Rule. — Extract the square-root of the quotient of the horse-power divided by the velocity in feet per second, and multiply it by .466.

Example. — The horse-power to be transmitted by a tooth is 60, and the velocity of it at its pitch-line is 6.66 feet per second; what should be the depth of the tooth?

$$\sqrt{\frac{60}{6.66}} \times .466 = 1.398 \text{ in.}$$

TO COMPUTE THE HORSE-POWER OF A TOOTH.

Rule. — Multiply the pressure at the pitch-line by its velocity in feet per minute, and divide the product by 33,000.

CALCULATING SPEED WHEN TIME IS NOT TAKEN INTO ACCOUNT.

Rule. — Divide the greater diameter, or number of teeth, by the lesser diameter or number of teeth, and the quotient is the number of revolutions the lesser will make, for one of the greater.

Example. — How many revolutions will a pinion of 20 teeth make, for 1 of a wheel with 125?

125 divided by 20 = 6.25 or $6\frac{1}{4}$ revolutions.

To find the number of revolutions of the last to one of the first, in a train of wheels and pinions: —

Rule. — Divide the product of all the teeth in the driving by the product of all the teeth in the driven; and the quotient equals the ratio of velocity required.

Example 1. — Required the ratio of velocity of the last, to 1 of the first, in the following train of wheels and pinions, viz.: pinions driving — the first of which contains 10 teeth, the second 15, and third 18. Wheels driven, first teeth 15, second 25, and third 32. $\frac{10 \times 15 \times 18}{15 \times 25 \times 32} = .225$ of a revolution the wheel will make to one of the pinion.

Example 2. — A wheel of 42 teeth giving motion to 1 of 12, on which shaft is a pulley of 21 inches diameter, driving 1 of 6;

required the number of revolutions of the last pulley to 1 of the first wheel. $\frac{42 \times 21}{12 \times 6} = 12.25$ or $12\frac{1}{4}$ revolutions.

NOTE. — Where increase or decrease of velocity is required to be communicated by wheel-work, it has been demonstrated that the number of teeth on each pinion should not be less than 1 to 6 of its wheel, unless there be some other important reason for a higher ratio.

WHEN TIME MUST BE REGARDED.

Rule. — Multiply the diameter or number of teeth in the driver by its velocity in any given time, and divide the product by the required velocity of the driven; the quotient equals the number of teeth or diameter of the driven, to produce the velocity required.

Example 1. — If a wheel containing 84 teeth makes 20 revolutions per minute, how many must another contain, to work in contact, and make 60 revolutions in the same time?

80×20 divided by $60 = 27$ teeth.

Example 2. — From a shaft making 45 revolutions per minute and with a pinion 9 inches diameter at the pitch-line, we wish to transmit motion at 15 revolutions per minute; what, at the pitch-line, must be the diameter of the wheel?

45×9 divided by $15 = 27$ inches.

Example 3. — Required the diameter of a pulley to make 16 revolutions in the same time as one of 24 inches making 36.

24×36 divided by $16 = 54$ inches.

The distance between the centers, and the velocities of two wheels being given, to find their proper diameters: —

Rule. — Divide the greatest velocity by the least; the quotient is the ratio of diameter the wheels must bear to each other. Hence, divide the distance between the centers by the ratio + 1; the quotient equals the radius of the smaller wheel; and subtract

the radius thus obtained from the distance between the centers; the remainder equals the radius of the other.

Example. — The distance of two shafts from center to center is 50 in. and the velocity of the one 25 revolutions per minute, the other is to make 80 at the same time; the proper diameters of the wheels at the pitch line are required.

80 divided by 25 = 3.2, ratio of velocity, and 50 divided by 3.2 + 1 = 11.9, the radius of the smaller wheel; then 50 minus 11.9 = 38.1, radius of larger; their diameters are 11.9 x 2 = 23.8 and 38.1 x 2 = 76.2 in.

To obtain or diminish an accumulated velocity by means of wheels and pinions, or wheels, pinions and pulleys, it is necessary that a proportional ratio of volocity should exist, and which is thus attained; multiply the given and required velocities together; and the square root of the product is the mean or proportionate velocity.

Example. — Let the given velocity of a wheel containing 54 teeth equal 16 revolutions per minute, and the given diameter of an intermediate pulley equal 25 in., to obtain a velocity of 81 revolutions in a machine; required the number of teeth in the intermediate wheel and diameter of the last pulley.

$\sqrt{81 \times 16} = 36$ mean velocity; 54 x 16 divided by 36 = 24 teeth, and 25 x 36 divided by 81 = 11.1 in., diameter of pulley.

TABLE OF THE WEIGHT OF A SQUARE FOOT OF SHEET IRON IN POUNDS AVOIRDUPOIS.

No. 1 is $\frac{5}{16}$ of an inch; No. 4, $\frac{1}{4}$; No. 11, $\frac{1}{8}$, etc.

No. on wire gauge,	1	2	3	4	5	6	7	8	9	10	11	12
Pounds avoir.,	12.5	12	11	10	9	8	7.5	7	6	5.68	5	4.62

No. on wire gauge,	13	14	15	16	17	18	19	20	21	22
Pounds avoir.,	4.31	4	3.95	3	2.5	2.18	1.93	1.62	1.5	1.37

SCREW-CUTTING.

In a lathe properly adapted, screws to any degree of pitch, or number of threads in a given length, may be cut by means of a leading screw of any given pitch, accompanied with change wheels and pinions; coarse pitches being effected generally by means of one wheel and one pinion with a carrier, or intermediate wheel, which cause no variation or change of motion to take place; hence, the following: —

Rule. — Divide the number of threads in a given length of the screw which is to be cut, by the number of threads in the same length of the leading screw attached to the lathe, and the quotient is the ratio that the wheel on the end of the screw must bear to that on the end of the lathe spindle.

Example. — Let it be required to cut a screw with 5 threads in an inch, the leading screw being of $\frac{1}{2}$ inch pitch, or containing 2 threads in an inch; what must be the ratio of wheels applied?

5 divided by $2 = 2.5$, the ratio they must bear to each other. Then suppose a pinion of 40 teeth be fixed upon for the spindle; $40 \times 2.5 = 100$ teeth for the wheel on the end of the screw.

But screws of a greater degree of fineness than about 8 threads in an inch are more conveniently cut by an additional wheel and pinion, because of the proper degree of velocity being more effectively attained, and these, on account of revolving upon a stud, are commonly designated the stud-wheels, or stud-wheel and pinion; but the mode of calculation and ratio of screw are the same as in the preceding rule. Hence, all that is further necessary is to fix upon any three wheels at pleasure, as those for the spindle and stud-wheels; then multiply the number of teeth in the spindle-wheel by the ratio of the screw and by the number of teeth in that wheel or pinion, which is in contact with the wheel on the end of the screw; divide the product by the stud-wheel in contact with the spindle-wheel, and the quotient is the number of teeth required in the wheel on the end of the leading screw.

Example. — Suppose a screw is required to be cut containing 25 threads in an inch, and the leading screw, as before, having two threads in an inch, and that a wheel of 60 teeth is fixed upon for the end of the spindle, 20 for the pinion in contact with the screw-wheel, and 100 for that in contact with the wheel on the end of the spindle; required the number of teeth in the wheel for the end of the leading screw.

$$25 \text{ divided by } 2 = 12.5, \text{ and } \frac{60 \times 12.5 \times 20}{100} = 150 \text{ teeth.}$$

Or suppose the spindle and screw wheels to be those fixed upon, also any one of the stud-wheels, to find the number of teeth in the other.

$$\frac{150 \times 100}{60 \times 12.5} = 20 \text{ teeth, or } \frac{60 \times 12.5 \times 20}{150} = 100 \text{ teeth.}$$

Transmission of Power by Manilla Rope. Horse-power Transmitted.

Feet per minute	1000	1500	2000	2500	3000	3500	4000	4500	5000
Diameter of Rope . . . $\frac{3}{4}$	$1\frac{3}{4}$	$2\frac{3}{4}$	$3\frac{1}{2}$	$4\frac{1}{2}$	$5\frac{1}{2}$	$6\frac{1}{2}$	7	8	9
" " . . . 1	$3\frac{1}{4}$	$4\frac{7}{8}$	$6\frac{1}{2}$	8	10	11	13	15	16
" " . . . $1\frac{1}{4}$	$5\frac{1}{8}$	$7\frac{1}{2}$	$10\frac{1}{4}$	13	15	18	20	23	26
" " . . . $1\frac{1}{2}$	$7\frac{1}{2}$	11	15	18	22	26	30	34	37
" " . . . $1\frac{3}{4}$	10	15	20	25	30	35	40	45	50
" " . . . 2	13	$19\frac{1}{2}$	26	33	39	46	52	59	65

Inches Expressed in Decimals of a Foot.

$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	1	2	3	4	5
.0208	.0417	.0626	.0833	.1667	.2500	.3333	.4167
6	7	8	9	10	11	12	
.5000	.5833	.6667	.7510	.8333	.9167	1.000	

TABLE OF TRANSMISSION OF POWER BY WIRE ROPES.

This table is based upon scientific calculations, careful observations and experience, and can be relied upon when the distance exceeds 100 feet. It is also found by experience that it is best to run the wire rope transmission at *the medium number of revolutions* indicated in the table, as it makes the best and smoothest running transmission. If more power is needed than is indicated at 80 to 100 revolutions, choose a larger diameter of sheave.

Diameter of Sheave in ft.	Number of Revolutions.	Diameter of Rope.	Horse-Power.	Diameter of Sheave in ft.	Number of Revolutions.	Diameter of Rope.	Horse-Power.
3	80	$\frac{3}{8}$	3	7	140	$\frac{9}{16}$	35
3	100	$\frac{3}{8}$	$3\frac{1}{2}$	8	80	$\frac{5}{8}$	26
3	120	$\frac{3}{8}$	4	8	100	$\frac{5}{8}$	32
3	140	$\frac{3}{8}$	$4\frac{1}{2}$	8	120	$\frac{5}{8}$	39
4	80	$\frac{3}{8}$	4	8	140	$\frac{5}{8}$	45
4	100	$\frac{3}{8}$	5	9	80	$\frac{9}{16}$ $\frac{5}{8}$	47 48
4	120	$\frac{3}{8}$	6	9	100	$\frac{9}{16}$ $\frac{5}{8}$	58 60
4	140	$\frac{3}{8}$	7	9	120	$\frac{9}{16}$ $\frac{5}{8}$	69 73
5	30	$\frac{7}{16}$	9	9	140	$\frac{9}{16}$ $\frac{5}{8}$	82 84
5	100	$\frac{7}{16}$	11	10	80	$\frac{5}{8}$ $\frac{11}{16}$	64 68
5	120	$\frac{7}{16}$	13	10	100	$\frac{5}{8}$ $\frac{11}{16}$	80 85
5	140	$\frac{7}{16}$	15	10	120	$\frac{5}{8}$ $\frac{11}{16}$	96 102
6	80	$\frac{1}{2}$	14	10	140	$\frac{5}{8}$ $\frac{11}{16}$	112 119
6	100	$\frac{1}{2}$	17	12	80	$\frac{11}{16}$ $\frac{3}{4}$	93 99
6	120	$\frac{1}{2}$	20	12	100	$\frac{11}{16}$ $\frac{3}{4}$	116 124
6	140	$\frac{1}{2}$	23	12	120	$\frac{11}{16}$ $\frac{3}{4}$	140 149
7	80	$\frac{9}{16}$	20	12	120	$\frac{7}{8}$	173
7	100	$\frac{9}{16}$	25	14	80	1 $1\frac{1}{8}$	141 148
7	120	$\frac{9}{16}$	30	14	100	1 $1\frac{1}{8}$	176 185

CHAPTER XXV.

ELECTRIC ELEVATORS.

In factories, warehouses and business buildings, freight, and in some instances passenger elevators, are operated by machines that are arranged to be driven by a belt. Such machines are variously called belted elevators, factory elevators and sometimes warehouse elevators.

In factories where there is a line of shafting kept running continuously, they are driven from it. As a rule the elevator machine is driven from a countershaft which latter is belted to the line shaft. Very often the elevator machine is driven directly from the line shaft. As the line shaft runs always in the same direction, the only way in which the elevator machine can be made to run in both directions is by the use of two belts, one open and the other crossed, or some form of gearing that will accomplish the same result. The common practice is to use double belts. Either one of these belts can be made to drive by using friction clutches, or by having tight and loose pulleys, and a belt shifter. The latter arrangement is the most common.

In buildings where there is no line of shafting, power for operating the elevator machine must be derived from some kind of motor installed expressly for the purpose. Nowadays electric motors are very extensively used for this purpose, and the combination of an elevator machine and an electric motor to drive it is very generally called an electric elevator, although in reality it is not such, but simply a belted elevator machine driven by an electric motor. It has become so common, however, to call such combinations electric elevators, that true electric elevators are generally designated as "direct connected electric elevators."

The first impression would be that in the combination of a belted elevator machine, and an electric motor to drive it, as the motor simply furnishes the power to set the machine in motion, there can be nothing about the combination that requires any special elucidation. Such a conclusion, however, would not be correct, for there are several ways in which the combination can be arranged, and in what follows we propose to explain these several combinations, pointing out the important features of each.

The simplest way in which a motor can be installed to drive an elevator, is to arrange it so as to drive the counter shaft continuously, in which case the elevator is stopped and started by throwing the belts on the tight or the loose pulley. Although this is a very simple arrangement, it is not desirable unless the elevator is kept in service all the time. In buildings where the elevator is used only at intervals, a great amount of power is wasted if the shafting is kept running all the time; hence it is desirable to arrange the motor so that it can be stopped when the elevator is stopped, and started whenever the elevator is to be used.

If the motor is arranged so as to run all the time, it is provided with a simple motor-starting switch, the same as is used for any motor installed to operate machinery of any kind. If the motor is started and stopped whenever the elevator is started and stopped, it is necessary to provide a motor-starter that can be operated from the elevator car. A very common way of arranging a motor to start and stop with the elevator is illustrated in the diagram Fig. 321.

In this diagram the elevator car is shown at *C*, with the lifting ropes running over the sheave *F* at the top of the elevator shaft, and then down and around the drum *A* of the elevator machine. This drum is driven by means of screw gearing, as a rule, with driving pulleys on the screw shaft as shown at *B*. The

driving motor is shown at M, and the counter-shaft to which it is belted is at D. In this arrangement the elevator machine is provided with a tight center pulley and loose pulleys on the two sides. The belts are shown on the loose pulleys, one being open and the

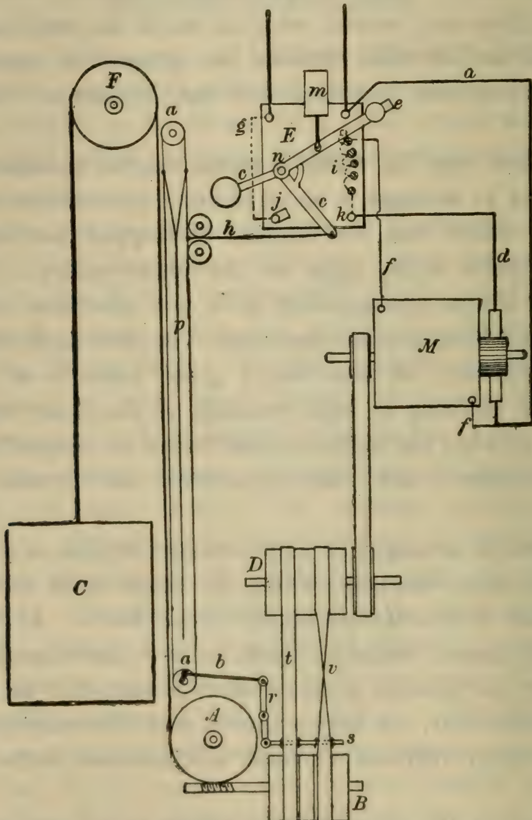


Fig. 321. Belt driven electric elevator.

other being crossed. The countershaft carries a drum wide enough to allow for the side movement of the belts when one or the other is shifted upon the tight center pulley by the belt shifter s. To operate the elevator car, a hand rope is provided which

runs up the elevator shaft at one side of the car from bottom to top of building. This rope is shown in the diagram at *l*, and runs around two small sheaves *a a*. The lower one of these sheaves is provided with a crank pin, which moves the connecting rod *b*, and thus rocks the lever *r*, and thereby moves the belt shifter *s*. To cause the car to ascend the hand rope *l* is pulled down, and to make the car descend, the hand rope is pulled up. As will be seen from this explanation, the lower sheave *a* will rotate in one direction when the hand rope is pulled to make the car go up, and in the opposite direction when the rope is pulled to make the car run down. In the diagram, sheave *a* is shown in the stop position, therefore when the hand rope is pulled down so as to make the car run up, the sheave will turn in a direction opposite to the movement of the hands of a clock, and thus the belt shifter will be moved to the right, and the open belt will be run onto the tight center pulley. If the hand rope is pulled up sheave *a* will rotate in the direction of the hands of a clock, and the belt shifter will move toward the left and thus shift the crossed belt onto the tight pulley. The rope *p* is a stop rope and is connected with the two sides of the hand rope in the manner shown, so that when the car is running in either direction, if *p* is pulled hard it will bring *l* to the position shown in the diagram, and thus stop the car. This rope can be dispensed with, but the objection is that in pulling the hand rope *l* to stop the car it may be pulled too far and then the car will not only be stopped but it will be caused to run in the opposite direction.

The motor starting switch is shown at *E*, the line wires being connected with the two top binding posts. The lever *c c* is in one piece and is independent of lever *e*, but both swing around the same pivot. At *m*, a dash pot is provided which acts to prevent the too rapid movement of lever *e*. As will be noticed, lever *c* has a projection which holds lever *e* up. The operation of this motor starter is as follows: When the hand rope *l* is pulled in

either direction, the rope h draws lever c towards the left and causes it to make contact with the switch jaw j. In this way the current from the upper binding post which is connected with j through wire g, passes to lever e, and thus to the starting resistance, which is indicated by the dotted lines i, to binding post k, from where it goes to the motor armature through wire d, and returns through the other wire d to the upper binding post at the right side, which is connected with the opposite side of the main line, thus completing the circuit. The field current branches off from the upper end of the starting resistance i and reaches the field coils through wire f, and through the lower wire f reaches the return armature wire d and thus the opposite side of the circuit. When the rope h pulls lever c over toward the left, the lever e does not follow it, as it is held up by the dash pot m. The weight on the end of e gradually overcomes the resistance of the dash pot, and thus causes lever e to move downward slowly. The velocity at which e moves downward is graduated by adjusting the opening in the dash pot through which the oil flows.

From the foregoing it will be seen that the starter E is made so as to accomplish automatically just what a man accomplishes when he moves the lever of an ordinary motor-starter; that is, it first closes the circuit through the motor, by bringing lever c into contact with j; and then allows lever e to move slowly so as to cut the resistance i out of the armature circuit gradually. When the elevator is stopped, by pulling the hand rope l to the stop position, the rope h slacks up and then the weight on the end of lever c causes it to descend, and thus return lever e to the position shown in the diagram, and also to break the circuit between c and j.

The elevator machine A is provided with a brake, which is actuated by the belt shifter s, so that when the belts are shifted upon the side pulleys, as shown in the diagram, the brake is put on, and thus the machine is stopped. As soon as the belt shifter

is moved to set the car in motion the brake is raised, so as to allow the machine to run free.

This arrangement is used very extensively, although the motor-starting switch is not always made in strict accordance with the one shown at E. In fact, there are a great many different designs on the market, but they all accomplish the same result, although the means employed may be very different.

Although it is very advantageous to have the motor arranged as in Fig. 321, so that it may be stopped and started together with the elevator, there is one objection to it which is sometimes regarded as serious, and that is, that as it requires a great amount of power to start an elevator from a state of rest, the motor will take a very strong current in the act of starting. To get around this objection, it is a common practice to provide a separate rope for starting the motor, and then when it is desired to use the elevator, the motor rope is pulled first, and in half a minute or so, the main hand rope is pulled. In this way the motor gets a start ahead of the elevator, and the headway of the motor armature helps to set the elevator car in motion, so that the current taken by the motor to start the elevator is very much reduced.

When a separate rope is used to start the motor in advance of the elevator, the starter E, or the levers connecting with it, are made so that while the motor can be started independently of the elevator car, when the main hand rope is pulled to stop the car, it also stops the motor. If this arrangement were not provided, the operator might stop the elevator and forget to stop the motor, in which case the latter would keep on running and waste power.

The main hand rope l is provided with stops at top and bottom of the elevator shaft, so that the car may be stopped automatically should the operator forget to pull the hand rope at the proper time.

It is the universal practice with elevator machines of the type shown in Fig. 321 to counterbalance the elevator car, but we have not shown a counterbalance in this diagram as it would only serve

to complicate its appearance, and it is not necessary to show it as the electrical features will be the same whether there is a counterbalance or not. This diagram also shows a separate rope *h* for actuating the starter *E*, but in actual machines *E* is generally

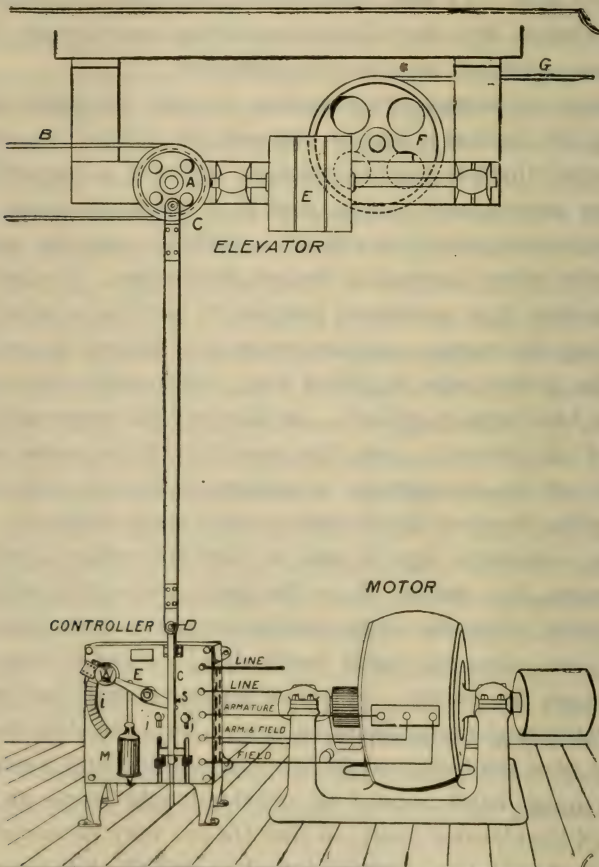


Fig. 322. Connections of gravity motor controller.

operated from the lower sheave *a*, which also actuates the belt shifter.

Fig. 322 is a diagram that shows the way in which one of the various motor starters in actual use is connected with the motor

and the operating hand rope. In this illustration A is the lower sheave a of Fig. 321, and F represents the hoisting drum and E the driving pulleys of the elevator machine, G being the lifting ropes from which the car is suspended. The sheave A is rotated

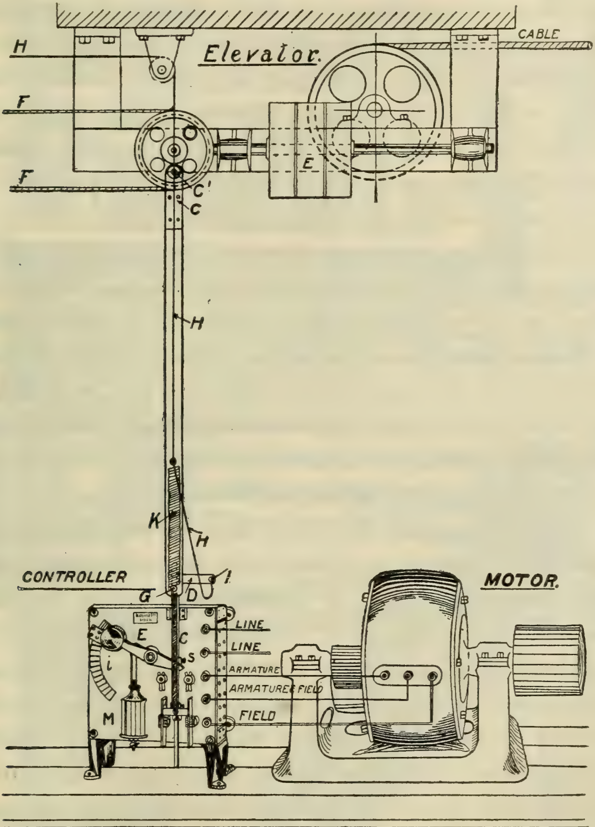


Fig. 323. Gravity controller with rope attachment.

through one quarter of a turn in either direction by the pull on the hand rope B, and when so rotated shifts the belt shifter and also lifts the brake from the brake-wheel. At the same time the crank pin C pulls up the connecting rod, and thus the upper end

of rod *c*, which takes the place of lever *c* in Fig. 321. In this way the switch blades in the lower end of *c* are raised into contact with the clips *jj*, which take the place of contact *j* in Fig. 321, and thus the circuit is closed. A projection *s* on *c* holds the switch *e* in the upper position, but when *c* is raised, *s* goes up with it, and then *e* is free to descend by the force of gravity acting upon the weight *w*. The dash pot *m* is set so as to retard the movement of *e* as much as may be desired. The outer end of *e* glides over the contacts *i* in its downward movement, and thus cuts out of the armature circuit the starting resistance. This resistance is contained in the controller box.

Fig. 323 shows the same type of controller as in Fig. 322, but it is arranged so that the motor may be started ahead of the elevator. The separate motor-starting rope is shown at *H*. When this rope is pulled, it elongates the spiral spring *K* which is connected with the stud *G* fixed in the upper end of rod *c*. The rope *H* is pulled up enough to stretch *K* until the lever *D* is lifted, *H* being attached to its outer end *l*. When *D* is lifted sufficiently, its inner end disengages the stud *G*, and allows it to slide upward in the slot shown in dotted lines, in the lower end of the connecting rod. In this way the motor is started ahead of the elevator machine. If now the elevator machine is started, by pulling on the main hand rope *FF*, the crank pin C' on the hand rope sheave will lift the connecting rod *C*, and when it reaches its upper position, the catch-lever *D* will drop into the position shown in the illustration, and thus lock the stud *G*, so that when the elevator is stopped, the rotation of the hand rope sheave will push rod *C* downward and thus stop the motor, as well as shift the belts and stop the elevator machine.

In the three illustrations shown the motor is run always in the same direction and the reversing of the direction of rotation of the hoisting drum is effected by the use of double belts and a belt shifter, or friction clutches, which cause one or the other of

the belts to do the driving. The way in which machines of this

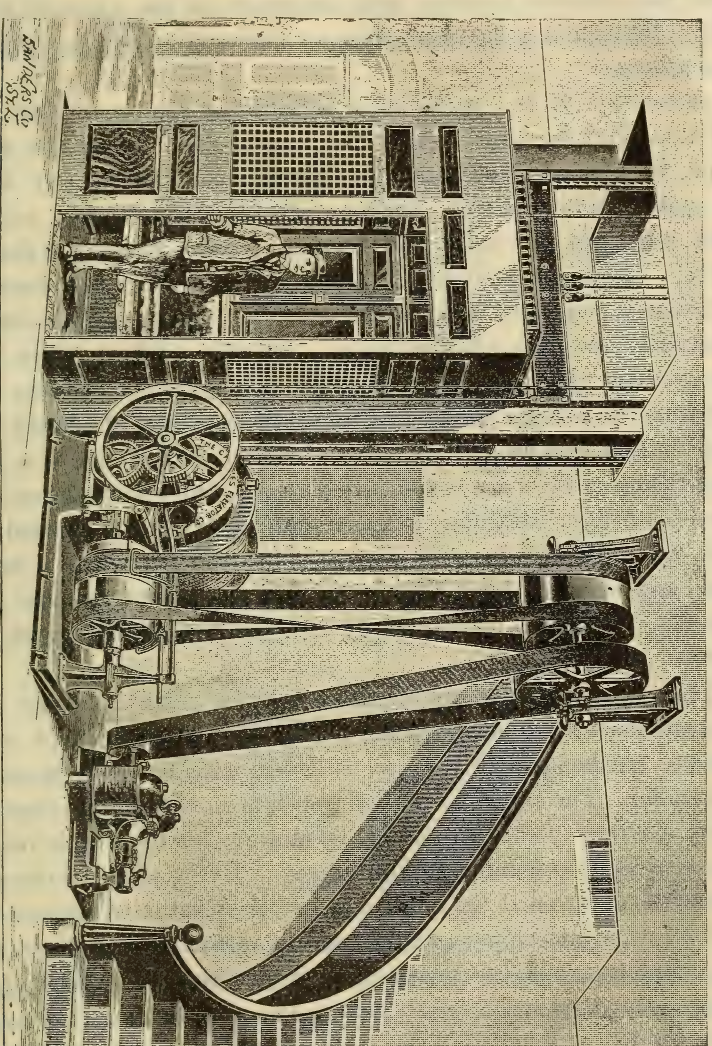


Fig. 324. Double belts to reverse direction of rotation of drum.

type are installed can be more fully understood from Fig. 324.

This figure shows the position of the motor, the countershaft and the elevator machine with reference to the elevator shaft. This illustration is so clear that an explanation of it would be superfluous.

In relation to the installation of elevator plants of this type all that need be said is that the motor must be of the shunt type, the same as those used for driving machines of any kind. A series wound motor, such as are used for electric railway cars, must not be used. Shunt wound motors cannot run above a certain speed, unless forced to do so by power applied from an external source, and in such an event they become generators of electricity and thus resist rotation. On this account, when they are used for elevator service, they not only move the elevator car, but when the latter is descending under the influence of a heavy load and tends to run away, the motor at once begins to act as a generator, and is thus converted into a brake, which holds the car and prevents it from attaining a speed much above the normal; in fact, the difference between the car velocity when lifting a heavy load, and when running down under the influence of a similar load is hardly enough to be noticed by any one not familiar with the elevator.

The motor in these combinations is to be given the same care as those used for other purposes; that is, it must be kept clean and the brushes properly set so as to run with as little spark as is possible. The controller switch requires more attention than the motor starters used with stationary motors, for the simple reason that it is used to a much greater extent. Every time the elevator is started or stopped the controller switch is actuated, hence, the switch levers are subjected to a considerable amount of wear, and the contacts are liable to become rough, either by cutting or by being burned on account of making imperfect contact. On this account the contact must be well examined at least once every day, and if burned or rough must be smoothed up. It is also

necessary to see that all parts of the controller are properly secured, that none of the screws or pins are working out, and that the contacts and switch levers are not out of their normal position.

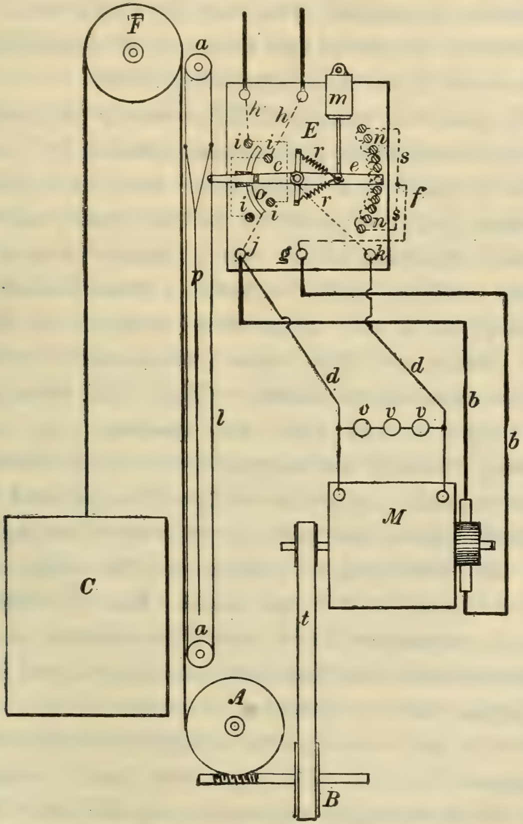


Fig. 325. Wiring used with reversible motor.

As electric motors can be run as well in one direction as the other, and as all that is required to make any motor reversible is to provide a reversing switch, it can be seen at once that by making use of such a switch, the direction of movement of the ele-

vator car can be reversed by simply reversing the motor, and thus do away with the complication of a countershaft and tight and loosé pulleys. Owing to this fact elevator machines are now made so as to be used with reversing motors. These are usually called single-belt machines. The way in which such machines are connected with the motor and the type of controller required can be understoood from the diagram Fig. 325.

As will be seen, the principal difference in the machine itself is that the tight and loose pulleys are replaced by a single tight pulley, which is only wide enough to carry the driving belt. Usually an extra pulley is provided for the brake, and this brake is mechanically operated in the same manner as upon machines provided with shifting belts. Another modification, which is sometimes used, but is not shown in the diagram, is the arrangement of a brake so that same is operated by a magnet instead of by mechanical means. With this arrangement the magnet is arranged so that when the machine is in motion, the current passing through the magnet coil acts to lift the brake, and when the machine stops, the magnet lets go, and the brake goes on. By arranging the brake in this way it becomes perfectly safe; for if the brake magnet fails to act, the brake will not be raised, and the machine will not move; that is, failure of the device to work properly will not permit the elevator car to move, thus calling attention to the fact that something is out of order.

The operation of the reversing controller is as follows: the current from the line wires passes along the dotted connections *h h* to the contact *i*,*i*, *i*,*i*. The upper left hand *i* contact is connected with the lower right hand one, and the upper right hand with the lower left hand. The switch lever *c* is connected with lever *e* by means of the two springs *r r*, so that *c* may be moved either up or down without carrying *e* with it. The curved contact *o* is connected with *j* and the stud around which *c* and *e* swing is connected with *k*, while *g* is connected with the ends of

the starting resistance *n n* by means of the wire *f* and the two wires *s s*. If the hand rope *l* is pulled so as to carry lever *c* upward, the current from the left side line wire will pass through upper left side *i* contact, to *o*, and thence to *j* and through wire *b* to the motor armature and returning through the other *b* wire will reach *g* and then pass through *f* and lower *s* to lower end of *n* and thence to lever *e* and the inner end of lever *c*, which will be resting on the upper right side *i* contact, thus reaching the right side line wire. The current for the field magnet coils will be drawn from *j* through wire *d* and back to *k* through the other wire *d*. As lever *c* has been moved upward, the upper spring *r* will be compressed, and the lower one will be stretched, hence a force will be exerted to move *e* downward over the lower contacts *n* and thus cut out the starting resistance. As in the case of the controller in Fig. 321 the dash pot *m* by its resistance retards the movements of *e*, so as to cut out the resistance as gradually as may be desired.

In the chapter on stationary motors it is shown that to prevent destructive sparking, when the starting switch is opened, the armature and field coils are connected so as to form a permanently closed loop. This style of connection is used in the non-reversing controller of Fig. 321, but it cannot be employed with a reversing controller, because both ends of the armature circuit must be free, so that they may be reversed when the direction of rotation is reversed. As this connection cannot be made, a very common expedient resorted to to prevent serious sparking when the switch is opened is to connect a string of incandescent lamps across the terminals of the field circuit, as is indicated at *v v v*. These lamps, together with the field coils, form a closed circuit, so that when the switch is opened, the field can discharge through the lamps, and thus avoid sparking at the controller contacts. The only objection to this arangement is that all the current that passes through the lamps is wasted, but by placing two or three

in series the loss is reduced to an insignificant amount. Another way in which the sparking is subdued, but only to a slight extent, is by connecting the brake magnet coil with the binding posts *j* and *k*, which is the simplest and most generally used connection. The brake magnet coil together with the field coils form a closed loop when connected with *j* and *k*, but when the main circuit is opened, the currents flowing in the two coils meet each other at *j* and *k* flowing in opposite directions, hence they both follow along the main circuit and try to jump across the gaps at the switch, and thus produce about as much sparking as if they were connected independently of each other. In tracing out the path of the current when lever *c* is moved upward, it was shown that the left side line went directly to the upper commutator brush. Now when *c* is moved downward, this same line wire runs to the lower commutator brush since the connections between the two upper *i* contacts and the two lower ones are crossed. To reverse the direction of rotation of a motor all that is required is to reverse the direction of the current through the armature, that through the field remaining unchanged, hence it will be seen that by crossing the connections between the upper and lower *i* contacts, the direction of rotation of the motor is reversed when the *c* lever is moved in opposite directions.

DIRECT CONNECTED ELECTRIC ELEVATORS.

The machines explained in the foregoing pages are simply combinations of an electric motor and a belt driven electric machine, but, as already stated, they are commonly spoken of as "electric elevators." In what follows it is proposed to explain the construction and operation of true electric elevators, which are called "direct connected machines" to distinguish them from the combinations so far described.

There are many designs of direct connected electric elevators

now upon the market, and it would be out of the question to undertake to describe all of them in the space that can be devoted to the subject in this book. On that account the discussion will be confined to the designs that are most extensively used. The explanations here given, however, will be sufficient to enable any

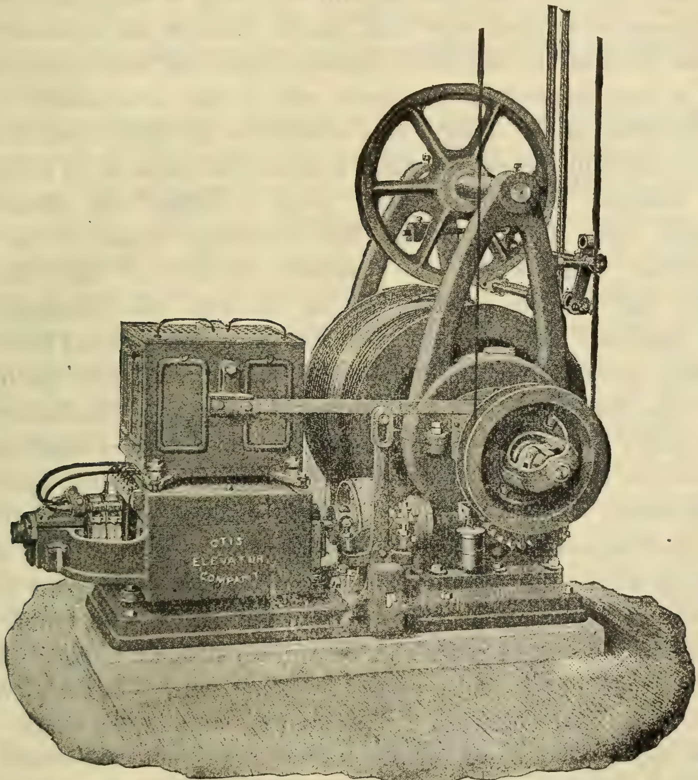


Fig. 326. The Otis direct connected elevator.

one to understand the operation of any of the machines not described because the difference in the principle of operation is only slight.

Perhaps the type of direct connected electric elevator that is most extensively used is the Otis drum elevator with hand rope control which is illustrated in Fig. 326. This machine has been upon the market for twelve years or more, and is still one of the standard Otis machines. It is called a hand rope control machine because the starting and stopping is controlled by the movement of a hand rope that passes through the elevator car. In the illustration, the sheave around which the hand rope passes can be seen located on the front end of the drum shaft. In a modification of the design, this sheave is mounted upon a seperate shaft but the way in which it acts is the same as in the present design. When the hand rope is pulled the sheave is rotated and the horizontal bar, running from it to the controller box, which is mounted on top of the motor, shifts the starting switch so as to run the machine in the direction desired. At the same time, the vertical lever extending upward from the side of the brake wheel, lifts the brake and thus frees the motor shaft so that it may revolve unobstructed. The motor carries a worm on the end of the armature shaft which gears into the under side of a worm wheel mounted upon the drum shaft. This worm wheel runs in a casing seen just back of the hand rope sheave wheel. The sheave mounted upon the shaft directly above the drum is for the purpose of guiding the counterbalance ropes, which run up from the back of the drum. In some buildings these ropes can be run up straight from the back of the drum, but in most cases they must run up in the elevator shaft in the space between the car and the side of the shaft. As these ropes wind upon the drum from one side to the other, the guiding sheave must move endwise on the shaft, hence it is called a traveling, or vibrating sheave. The levers seen projecting to the right of the machine from a small shaft just above the drum are what is called a slack cable stop, and their office is to stop the machine if the lifting cable becomes slack through the wedg-

ing of the car in the elevator shaft or any other cause. These levers are held in the position shown when the lifting ropes are tight, but drop out of position if the rope slackens up, and in dropping they release a lever, which holds the weight seen under the hand rope sheave. The movement of this lever operates a catch that engages with the hand rope sheave and thus the horizontal bar that operates the brake and the controller switch is brought to the stop position and the rotation of the hoisting drum is stopped.

The hand rope has fastened to it at the top and bottom of the elevator shaft stops that are moved by the car when it reaches either end of its travel, and thus the elevator machine is stopped automatically. This arrangement is the same as that used with the belt driven machines already described, but as an additional safety, a stop motion is provided on the machine itself, so that if the stops on the hand rope become displaced, the car will still be stopped automatically at the top and bottom landings. This stop motion is seen on the end of the shaft, just in front of the hand rope sheave, and consists of a nut that travels on the shaft as the latter revolves. At both sides of the screw there are projection cases upon the inclosing frame, which are struck by the traveling nut when it comes near enough to either end. When the nut strikes the projection, the hand rope sheave is revolved with the shaft and thus the machine is stopped. To understand this action it must be remembered that the hand rope sheave does not revolve except when turned by the pull on the hand rope or by the action of the slack cable stop or the traveling nut.

The controller box on top of the motor contains the starting resistance, the starting and reversing switch, and also a magnet to actuate a switch that gradually cuts out the starting resistance. The way in which the switches act to start and stop the motor can be readily explained by the aid of the diagram Fig. 327.

This shows the circuit connections in the simplest possible

form. In this diagram all the wires whose presence would make

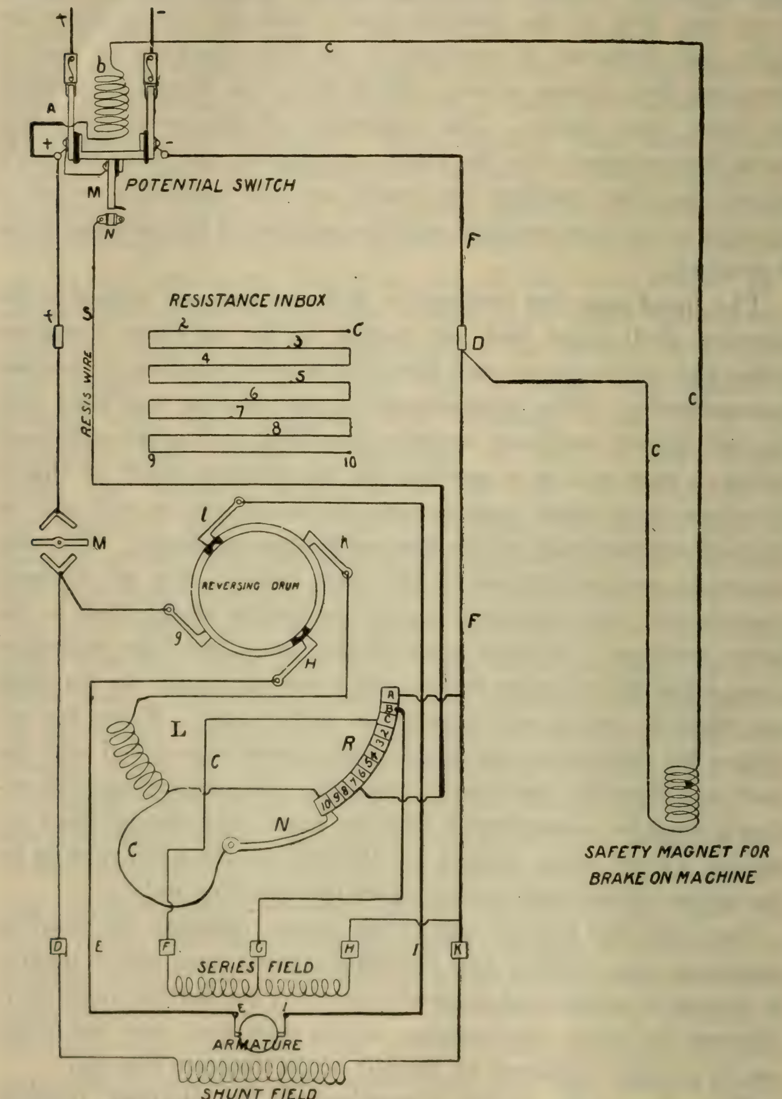


Fig. 327. Diagram of controller box and wiring.

the drawing confusing have been removed, but the manner in

which they are connected will be readily understood from the following explanation: —

The main switch, which connects the motor circuits with the line, is located at the upper left hand corner of the diagram, the main line wires being marke + and — . When this switch is closed, the motor circuits are connected with the line, but the motor circuit itself is not closed so long as the switch *M* remains in the position shown. When this switch is turned about one quarter of a revolution in either direction, one end will ride over the upper contact and the other one over the lower contact. The reversing drum and switch *M* are mounted on the same spindle and move together. They are located within the controller box, on top of the motor, and are moved by the horizontal bar; see Fig. 326. The shaded portions of the drum, on which the brushes *h* and *i* rest are made of insulating material so that when switch *M* and the reversing drum are in the position shown the motor circuit is open at two points. This is the position of these parts when the machine is stopped.

The starting resistance is shown above the reversing drum, and in the machine it occupies the space at the back of the controller box, shown on top of the motor in Fig. 326. The segment *R* is a series of contacts that are connected with the resistance in the resistance box; No. 2 contact being connected with point 2 on the resistance and so on for all the other numbers. The switch arm *N* is moved over the contacts *R* by a magnet that is represented by the spiral *L*. The motor armature and the shunt and series field coils are shown at the bottom of the diagram. The motor is compound wound, it being made so for the purpose of keeping the starting current as low as possible. The path of the current through the wires is as follows: Suppose the reversing drum and the *M* switch are revolved in the direction in which the hands of a clock move, then brushes *g* and *i* will rest on one segment, and *h* and *k* will rest on the other segment.

As switch M will now be closed, the current will flow to brush g and through the reversing drum segment to brush i; then it will follow the wire to the right side I of the armature and passing through the latter will reach wire E and thus brush h, from which it will pass to brush k. From this brush the current will go to and through magnet L and by wire C' and switch N will reach contact No. 10. As this contact is connected with point 10 of the resistance the current will reach the latter and will pass through the whole of it, coming out at the opposite end C. This end is connected with contact C, so that from this segment the current can flow through wire C to the end F of the series field coils, and passing through these to end H, will find its way to wire f, and thus return to the opposite side of the main line. From this explanation it will be seen that the current will pass through the motor armature, and then through the whole of the resistance in the resistance box, and then through the series field coils, and finally reach the other side of the main line. From the switch M another current will branch off and run to binding post D, and thence through the shunt field coil to binding post K and thus to wire f, and through the latter to the opposite of the main line.

The switch lever N is in some cases arranged so that the magnet L acts to hold it upon contact 10 and a spring acts to carry it forward toward contact A; in other cases the magnet is wound with two coils, one of which pulls N in one direction and the other pulls it in the opposite direction, the two coils being so proportioned that N moves gradually from contact 10 toward contact A. If we take the spring arrangement, then magnet L will pull N back toward contact 10, and the spring will pull it forward. As the starting current is very strong, N will be held on contact 10, but as the current weakens, the spring will begin to overpower the magnet, and N will slide over contact 9 and then 8 and 7 and so on to contact A. As contact 9 is connected with the point 9

of the resistance, when N reaches it, the section of the resistance between points 10 and 9 will be cut out. When N reaches contact 7 the resistance between points 10 and 7 will be cut out for the latter point is connected with contact 7. As all the contacts are connected with the corresponding points of the resistance, when N reaches contact C, all the resistance in the resistance box will be cut out of the circuit. As will be noticed, contact B is connected with the center point G of the series field coil so that when N reaches contact B one-half of the series coils will be cut out in addition to the whole of the resistance box. When N reaches contact A the current will pass directly to wire f, and thus cut out all the series field coils and then the motor will run as a plain shunt-wound machine, and its speed will be the highest it can attain.

If the reversing drum and switch M are now revolved to the position shown in the diagram, the circuit through the motor will be broken and the machine will come to a state of rest. If the reversing drum and M are now revolved in the opposite direction, that is, contrary to the movement of the hands of a clock, the brushes g and h will rest on one of the revolving drum segments, and i and k on the other segment. If the path of the current is now traced it will be found that it will enter the armature through wire E, and the left side, instead of through wire I, as in the previous case. It will also be found, however, that the current after passing through the armature will reach the series field coils through F, which is the same path as before, so that the direction of the current has been reversed through the armature only, which is what is required to reverse the direction of rotation of the motor. Whichever way the switch M and the reversing drum are turned, the direction of the currents through the series field coils and the shunt field coil will be the same, and only the armature current will be reversed.

Cutting out the series field coils not only increases the speed

of the motor, but obviates the danger of the car attaining a dangerously high speed if the load is being lowered. A shunt wound motor will run as a motor up to a certain speed, but if the velocity is forced above this point by driving the machine by the application of external power, then the motor will begin to act as a generator, and as it takes power to run a generator the motor will begin to hold back. Now if an elevator car is running down with a heavy load, the load will draw the car down, and unless a resistance of some kind is interposed, the speed will become greater and greater as the car descends, and by the time it reaches the bottom of the shaft it may be running at a velocity almost equal to that attained by a free fall. The power required to drive the motor when acting as a generator serves to hold the car back, for the current developed increases very rapidly with increase of speed, so that an increase of speed of ten or fifteen per cent above the normal running velocity will be about as much as can be reached even with an extra heavy load.

Although the motor will act as a generator and hold the car so that it cannot attain a dangerous speed when descending under the influence of a heavy load, it will only accomplish this result when the circuit is closed; for if the circuit is open there will be no power generated; hence, no power will be absorbed by the motor. As can be readily seen, it is possible for the circuit outside of the motor to become broken by the melting of a fuse or some other cause, and if this occurs when the car is coming down with a heavy load there might be a serious accident. To obviate such mishaps the main switch is made with a magnet b, which holds the switch closed so long as current passes through it, but allows the switch to swing open if the line current disappears. This switch on this account is called a potential switch, because it is arranged to be actuated by the difference of potential between the two sides of the line. When the line current fails, and the potential switch opens, the blade m comes into contact with n

and thus the circuit for the motor armature is closed through the resistance wire *s*, which is connected with contact 7. This connection short circuits the armature through a resistance sufficient to keep it from being burned out, but not enough to prevent the motor from acting as a brake and holding the car down to a safe speed.

The wire *c c*, which runs from magnet *b* of the potental switch, it will be noticed, connects with a coil marked safety brake magnet. This magnet acts normally to hold the brake off when the machine is running, but if the current passing through it dies out, then it acts to put the brake on. Now, as has already been explained, when the current is flowing in the main line, there is a current passing through coil *b* of the potential switch; hence, there is a current passing through the coil of the safety magnet for the brake; but if the line current fails the current through the brake magnet will also fail and the brake will go on; so that the car will be doubly protected, one protection being the short circuiting of the motor circuit through wire *s*, and the other the applying of the brake by reason of the failure of the current to flow through the safety brake magnet.

As to directions for the proper care of these machines, very little need be said, as they are simple and substantial in construction and give very little trouble. The motor proper requires the same attention as is given to any stationary motor, that is, the commutator and all other parts must be kept as clean as possible and the brushes must be properly set. As to the other parts, all that need be said is that the bearings must be well lubricated and free from grit. They must be tight enough to not allow the parts to play, but at the same time care must be taken that they are not so tight as to heat up or cut. All bolts and nuts must be regularly examined and kept tight, so that they may not work loose or out of place. The most important point to observe, however, is not to undertake under any circumstances

to tinker with the sheave wheel and the gears that connect it with the horizontal bar that operates the brake and controller switches. Neither must the brake or the switches be disturbed. All that is to be done to the latter is to keep the contacts bright and clean. If any of these parts, from the sheave wheel to the controller switches, get out of set, so that the machine will not run satisfactorily, do not undertake to readjust them, but send for an expert from the elevator company. If any of these parts are removed or shifted there is danger of their not being put back in their proper position, and if they are misplaced a very serious accident may be the result. If the proper adjustment of these parts is destroyed, the elevator will not stop automatically at the top and bottom landings, but will run too far at one end and stop short of the mark at the other; hence, the car may either strike violently against the floor or run at full speed into the overhead beams, and in either case the results might be very serious. Even elevator experts have to go cautiously in adjusting the position of the sheave wheel and the parts connected with it.

The fact that those not thoroughly posted in the operation of these elevators should not tamper with the hand rope sheave and its connections, is not at all unfortunate, for it is next to impossible for them to get out of place; but special caution is advised at this point, because there are many men who are apt to take it for granted that if the machine runs poorly from some trifling cause that they have not been able to locate, the trouble must be due to some defect in the adjustment of the several parts of the operating sheave and its connections. They will then proceed to pull the machine apart, and when they put it together again they are very liable to get it connected wrong, and if such should be the case the first trip made by the elevator might end seriously.

Although the machine described in the foregoing works in an entirely satisfactory manner, it has been superseded almost en-

tirely in first-class installations of recent date by machines that are controlled by means of a small switch in the car instead of the hand rope. There are several types of such elevators made by the Otis company, one of the latest designs being shown in Fig. 328.

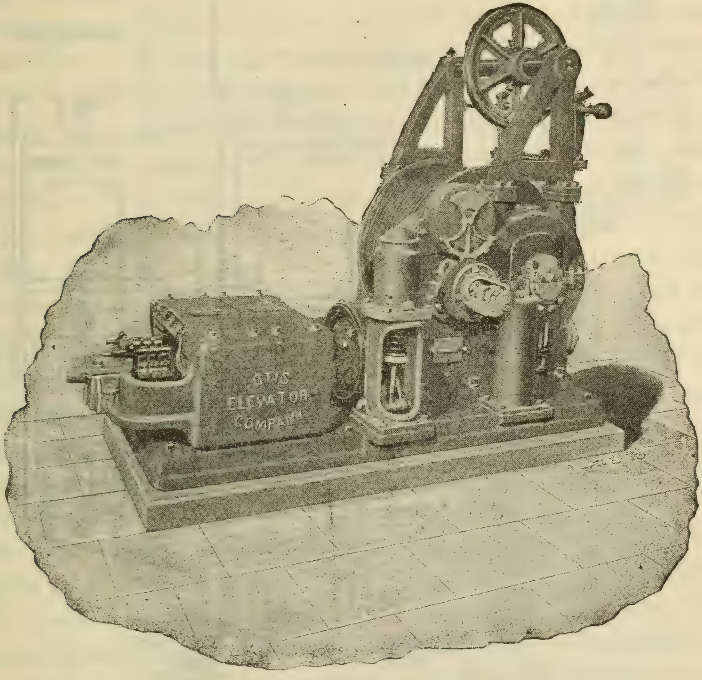


Fig. 328. Latest design of direct connected machine.

As will be noticed at once, this machine is different in several respects from the hand rope control machine shown in Fig. 326. As the machine is controlled by the movement of a switch in the car, the brake cannot very well be actuated mechanically, hence a magnetic brake is provided, the magnet being seen at the top of the stand to the right of the motor. The automatic stopping devices and the slack cable stop are also arranged so as to act upon

switches, which are contained within the casings seen at the front end of the hoisting drum. The controller for this type of machine is not placed on top of the motor, generally, for since it

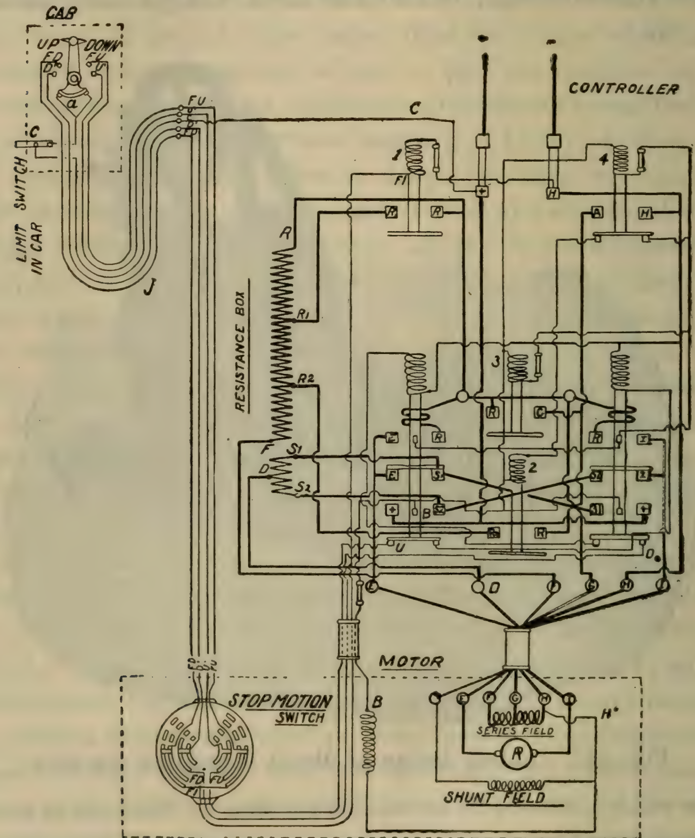


Fig. 329. Diagram of wiring connections for controller.

is not connected mechanically with any of the moving parts of the machine, it can be located at any convenient point, and is then connected with the motor armature, field coils and with the brake

magnet and automatic stop switches by means of copper wires. The controller used with this type of machine is arranged after the fashion of a switchboard, the switches being located on the front, and the connecting wires, together with the starting resistance, being at the back. The switches are actuated by means of electromagnets, and on that account the device is called a magnet controller. The diagram of the wiring connections with this controller is more complicated than that for the hand rope controller, but for the purpose of simplifying the drawing as much as possible we have removed all the connections that are not actually necessary for a proper understanding of the general arrangement of the circuits. This simplified diagram is shown in Fig. 329.

The front of the controller is shown in Fig. 330, and the back of same in Fig. 331, the starting resistance being removed in this illustration so as to afford a clear view of the wire connections. The side of the starting resistance can be seen in Fig. 330. In this last named illustration, all the switches are in the position they take when the elevator is stopped. The two large switches on either side at the bottom of the board are the starting switches, one acting to run the car up and the other one to run it down. The two smaller switches occupying the center of the bottom panel of the board and the two switches in the upper corner are for the purpose of accelerating the velocity of the motor when it is started. When the motor starts, there is a resistance in the armature circuit, and the current after passing through the armature is passed through series field coils. After the motor has started, the starting resistance is cut out, and then the series field coils are cut out, so that when the full speed is attained, the motor is a simple shunt-wound machine. In this respect the arrangement of the motor circuits is the same as in the hand rope controller machine.

When it is desired to start the car, a small switch in the latter is moved toward the right or left, according to the direction in

which the car is to move. To run the car up, the car switch is turned to the left, and this movement sends a current through the magnet of the lower right side magnet on the controller board.

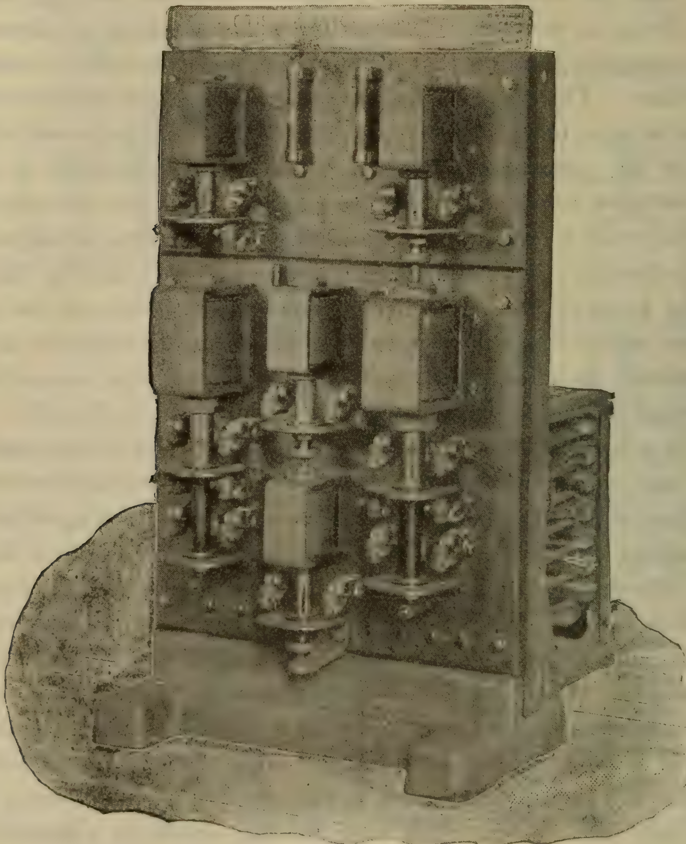


Fig. 330. Showing front of controller.

This magnet then lifts its plunger and the two discs mounted upon the latter come into contact with the stationary connectors located just above them, and then the current can find its way through

the motor circuits in the proper direction to produce the upward motion. The four small switch magnets on the controller board are connected in separate circuits that are in parallel with each

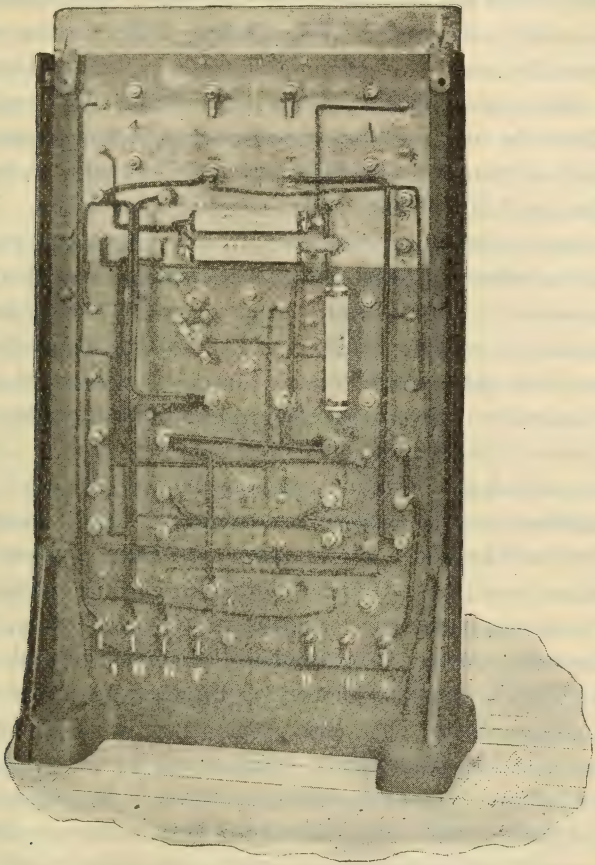


Fig. 331. Back of controller.

other, and in shunt relation to the armature of the motor. When the motor first starts, the counter electromotive force developed by the armature is not as great as when it is running at full speed,

because a portion of the electromotive force of the line current is used to force the current through the starting resistance and through the series field coils. When a portion of the starting resistance is cut out the armature counter electromotive force is correspondingly increased. When more of the starting resistance is cut out, the counter electromotive force is further increased.

It is still further increased when the series field coils are cut out. Now the current that passes through the magnets of the four small switches on the controller board increases as the counter electromotive force of the motor armature increases. The magnets are so adjusted that as the currents passing through them increase one after the other will lift its plunger and then the connections made by the discs at the lower end of these plungers will cut out successively the sections of the starting resistance and the sections of the series field coils. The two small tubes at the top of the controller board are safety fuses, and the line wires are connected with their upper ends.

By the aid of the foregoing explanation of the way in which the controller acts, the following description of the wiring diagram (Fig. 329) will be easily understood. In this diagram the line wires come in at the top of the controller and are marked + and —. The motor is shown at the bottom of the diagram, the circle *A* representing the armature, and the coil *B* is the brake magnet. The stop motion switch is placed on the elevator machine, in one of the casings at the front end of the drum, and is actuated by the automatic stop mechanism, which stops the car at the top and bottom landings. The car switch is shown in the upper left hand corner of the diagram, and the curved lines *J* represent the wires that connect it with the motor and the controller board. These wires are placed within a flexible cable that is attached to the side of the elevator shaft half the way up from the bottom, the cable being long enough to reach the car when at either end of the shaft. The limit switch in the car is for the purpose of stop-

ping the motor, if the car reaches either end of its travel without being stopped by the operator, or the action of the stop motion switch. This switch is closed under ordinary conditions, so that the current in wire C can flow all the way to the lower contact a of the car switch. If it is desired to run the car down, the car switch is turned to the right, and then wire C is connected with wires D' and FD. The stop motion switch is normally in the position shown so that the current in wire D' can pass to D_0 and following this wire it will reach contact D_0 which is under the lower disc of the right side starting switch. Through the disc this contact is connected with the corresponding contact on the other side of the disc, and this latter contact is connected with a wire that carries the current to the magnet of the left side starting switch. Considering now the main current in the + line it can be seen that it can flow down to the line near the bottom of the controller portion of the diagram, and which terminated in the + contacts of both the starting switches, but can go no further so long as the discs on the plungers of the magnets are in the lower position. As soon, however, as the current coming from the car switch passes through the magnet of the left side switch, as just explained, the plunger will be lifted, and then the disc will connect the + contact with the $S2$ contact, and also with a smaller contact B. When this connection is made, the main current can flow from contact $S2$ to contact $S2$ of the right side switch, and thence through the connecting disc to contact l which is connected by wire to binding post l; the latter being connected with the right side armature terminal l. After passing through the armature the main current reaches binding post E and through the connecting wire the contact E at the top of the left side starting switch, and as the plunger of this switch is in the raised position, the current can pass to contact R and thus reach the upper end R of the starting resistance in the resistance box.

From the end F of the starting resistance, the main current flows to binding post F and then to the F end of the series field coils, and from end H to binding post H and to the — line wire at the top of the diagram. The current for the shunt field is taken from the contact $S2$ at the bottom of the left-side starting switch, and passes to point $S4$ and thence to D' and to the D end of the shunt field coil, and through this coil to end H of the series coil, and thus to the — line. The current for the brake magnet starts from the small contact B at the bottom of the left-side starting switch.

The car switch when moved will first cover contact D' so that the main current will follow the path outlined above, but as soon as the car switch covers contact FD, the current passing through wire FD in the cable will reach the stop-motion switch and pass to F, and thus to magnet No. 1 at the upper left hand corner of the controller board. The lifting of this switch will cause its disc to connect the contacts RR' and thus the current will pass to point R' of the resistance and cut out the upper section. The current from contact B at the bottom of the left-side starting switch passes through the magnet coils of the three switches, Nos. 2, 3 and 4. Now soon after the first section of the starting resistance is cut out, No. 2 magnet becomes strong enough to lift its plunger, and then the current from the right side, contact R, at the top of the left-side switch, will pass to contact R of No. 2 switch, and thus to $R2$ and to point $R2$ of the resistance, thereby cutting out two sections. In this way the current through magnet of switch No. 3 will be increased and the plunger will be lifted so that the current will be able to pass from the R contact of this switch to the G contact, and thus to binding post G and to the center of the series field coils, thereby cutting out one-half of these coils. In this way the current through coil of No. 4 magnet will be further increased, so that it will be able to lift its plunger, and thus form a direct connection from contact G of switch No. 3 and the main wire leading to the — line.

Thus it will be seen that the four switches, 1, 2, 3 and 4, will act one after the other. This same operation is repeated if the car switch is moved to the right, so as to run the elevator down, the only difference being that the starting switch at the right side of the board will be lifted, but the action of the four smaller switches will be the same.

In addition to the operating circuits described in the foregoing there are wires that connect the slack cable switch with the motor circuits and other connections by means of which the elevator may be run from the controller board whenever desired. These connections are not shown in Fig. 329, as they would complicate the drawing, and it is not thought advisable to complicate the explanation of the main part of the system for the sake of introducing the minor details.

This type of electric control is used for elevator machines installed in office buildings, and others placed where the car is operated by a regular attendant. For private house elevators and for dumb waiters it is necessary to modify the controlling system so that the car may be operated not only from within, but also from any of the floors of the building. It is further necessary that the circuit connections be such that if the car is operated from any floor, it will run to that floor, whether above or below it, and further, so that if it is being operated by a person within the car it cannot be operated by any one else from any of the landings. It must also be arranged so that if the car is set in motion from any floor it cannot be stopped or interfered with in any way by a person at another floor. For the purpose of safety the system must also be arranged so that the car cannot move away from any floor until the landing door is closed. This feature is necessary to guard against people falling through the open doorway into the elevator shaft. Although it would appear difficult to accomplish all these results without resorting to great complications, as a matter of fact the system used by the Otis company is decidedly

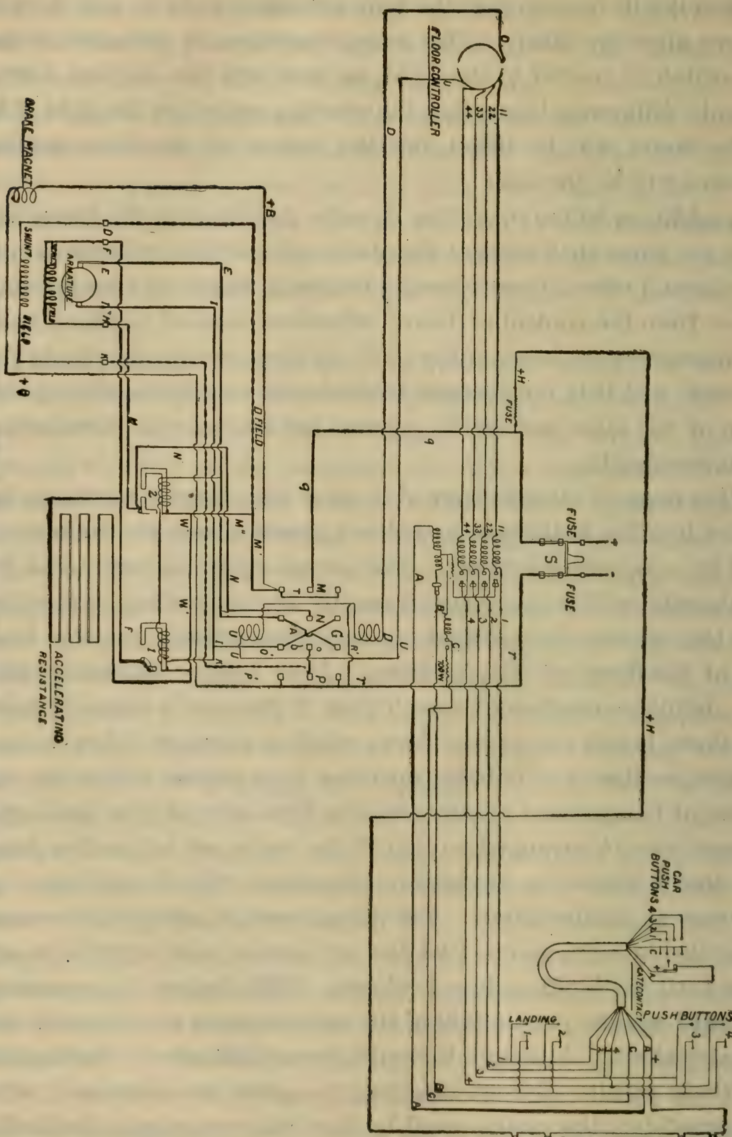


Fig. 332. Simplified diagram of elevator controller.

simple. At each floor of the building a push button is placed, and by pressing this for an instant the car is set in motion wherever it may be, providing it is not being used by some other person, and when it reaches the floor from which it has been operated it will stop automatically. If the elevator is operated from the car, a button is pushed that corresponds to the floor at which it is desired to stop, the car will then begin to move, and when the floor is reached it will stop. If the passenger after stepping out of the car forgets to close the landing door, the elevator cannot be moved away from the landing by the manipulation of any of the push buttons on the various floors or within the car. The way in which all these results are accomplished can be made clear by the aid of Fig. 332, which is a simplified diagram of the wiring.

In this diagram most of the parts are marked with their full name. The floor controller is a drum, which is revolved by the elevator machine, and its office is to shift the connections of the wires 11, 22, 33, 44, from one side of the circuit *DU* to the other as the car ascends and descends in the elevator shaft. This shifting of these connections is necessary to cause the car to run down if above the landing from which it is operated, and to run up if it is below the landing. The actual position of the floor controller with reference to the elevator machine can be seen in Fig. 333 in which the floor controller is located back of the motor and is driven from the drum shaft by means of a chain and sprocket wheel. In the diagram Fig. 332 it will be noticed that the drum surface is divided into two segments and upon one rests the brush of wire *D* while upon the other rests the brush of wire *U*. The twelve contacts shown at *G* form the operating switch. The center row marked *m n o p* are movable, and the four contacts above them as well as the four below are stationary. The center row of contacts *m n o p* are moved upward by a magnet represented by the coil *D* and they are moved downward by another magnet repre-

sented by the coil U. From this it will be seen that if a current comes from the floor controller through wire D the movable contacts of G will be lifted and will connect with the top row, while if the current comes from the floor controller through wire U, the movable contacts will be depressed and will make connections with the lower row of contacts.

The main switch that connects the motor circuits with the main line is shown at S. As will be noticed, a wire marked $d + H$ runs from the $+$ wire to the right side of the diagram, where the landing and the car push buttons and their connections are shown. This wire runs from top to bottom of the elevator shaft and is connected with switches that are closed when the landing doors are closed, and open when the doors are open. These switches are indicated by the four circles marked door contacts, the diagram being for a building four stories high. If the door contacts are closed, the current can pass as far as the wire marked $+$ which runs through the flexible cable to the car. In the car there is a switch in this wire and further on a gate contact, which is closed when the car door is closed. If these switches are closed, the current can return from the car through wire A and run as far as the center of the diagram under the main switch S. The floor controller is shown in the position corresponding to the car at the bottom of the shaft. Suppose now that the landing push button l is pressed for a second, then the wires B and l will be connected, and the current in wire A will pass to wire B and through the push button to wire l and thence to wire ll. The coil between wire l and wire ll is a magnet, and as soon as the current passes through it, it draws the contact to the right and thus provides a path for the current direct from wire A to wire ll, so that the push button may be raised without opening the circuit. The current in wire ll will pass through the floor controller to wire U and thus through magnet U of the operating switch G. This magnet will then draw down the movable contacts m n o p, and the main line

current from the + wire will pass from contact m to wire m' and through wire m' to point w, hence through wire w' to the accelerating, or starting resistance, and to wire F which leads to the series field coils. Returning from these coils through wire H to magnet switch 2 and thence wire n' to contact n, and as this con-

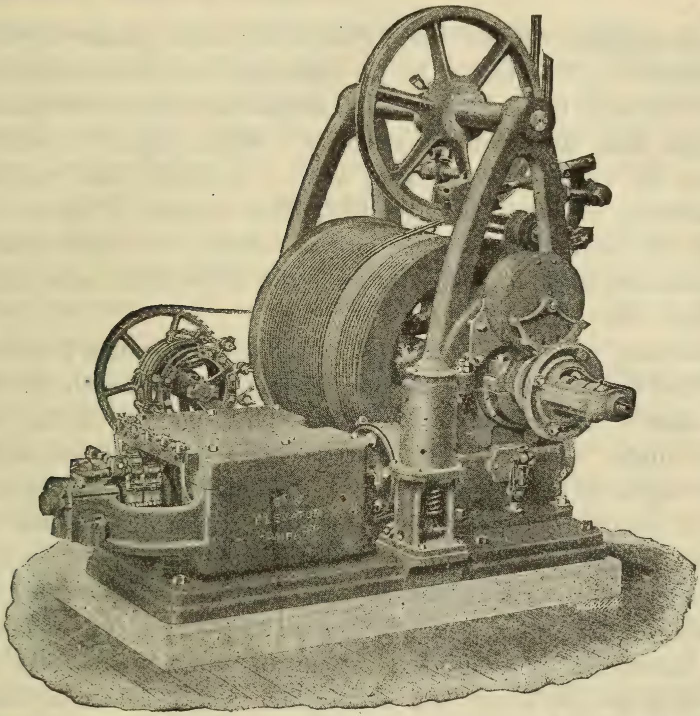


Fig. 333. Direct connected elevator with floor controller.

tact is pressing against the one directly below it, the current will flow through the connection to wire E and thus to the armature; returning from the latter through wire I and wire o' to the contact below o and thus to o and through the permanent connection to contact p and to the lower right hand contact, which is connected

with wire r, which runs to the — side of the main switch. The shunt field current is derived from wire m' and returns to contact p and thus to wire r through wire p', as can be clearly traced. The brake magnet current starts from the left side contact of G through wire $+B$ and returns directly to the lower end of wire r.

The magnet switches 1 and 2 act in the same manner as those in diagram Fig. 329, that is, by the increase in the counter electromotive force of the armature which causes the current that passes through them to increase in strength. When magnet l closes its switch, the current passes from wire w' to wire F and thus the accelerating resistance is cut out. When magnet 2 closes its switch the current passes from wire m'' directly to n' and thus to the armature without going through the series field coils; thus the latter are cut out.

Returning now to the operation of the floor controller it will be seen that as the current is flowing through wire ll the circuit will be broken if the controller is rotated until the gap at the top comes under the brush of wire ll. Now the floor controller drum begins to turn as soon as the elevator machine moves, and it is so geared to the elevator drum that when the car comes opposite the first floor the brush of wire ll will be over the upper gap, and then the circuit will be open and the magnet U will be de-energized and allow switch G to move back to the stop position.

If button No. 4 is pressed instead of No. 1 the car will not stop until the gap at the top of the floor controller drum comes under the brush wire 44, for the circuit between this wire and wire U will be closed until that position is reached.

If the car is run up to the fourth floor, as the gap at the top of the floor controller drum will then be under the brush of wire 44, the brushes of wire 11, 22 and 33 will rest upon the same segment as the brush of wire D; therefore, if with the car at the top floor a button is pressed at any one of the lower floors

the current will pass from its corresponding wire to wire D and thus through magnet coil D and to wire r' and wire r. The current passing through magnet D will draw the movable contacts of the operating switch 6 upward, and thus set the elevator machine in motion in the opposite direction from that in which it runs when the U magnet is energized.

In tracing out the circuits from the floor push buttons as just explained it will be noticed that if any one of them is depressed, the current in wire A will flow through wire B to the button depressed, and then enter the wire returning from that button. When the car buttons are depressed the current in wire A will pass to wire C and then through the button in the car to the proper return wire; that is, to one or the other of the wires 1, 2, 3, 4. After entering one of these four wires the current follows the same path as it does when one of the floor buttons is depressed. The magnet B' in the B wire, and the magnet C' in the C wire, are for the purpose of preventing interference between a person operating the elevator from within the car and another one at one of the landings. The B' switch is actuated by a magnet that is wound with two coils that act in opposition to each other. These coils are shown to the left of B'. When the elevator is operated from one of the floor push buttons the current in wire A passes through both the coils on the magnet of switch B' and as one coil counteracts the other the switch is left closed and the current passes directly to wire B. If the elevator is operated from within the car the current from wire A in passing to wire C passes through one of the coils of the magnet that actuates switch B', hence this switch is opened and the connection with wire B is broken, so that if now any one of the floor buttons is pressed it will have no effect as the circuit is opened at switch B'. The current flowing through wire C passes through a magnet that acts to close the switch C' and thus allow a portion of the current to pass directly to wire r. This current

will continue to flow even after the car has stopped at the landing, providing the door is not opened. As soon as the door in the car, or the landing door, is opened the circuit is broken either in wire *H* or in wire *A*, and then the car cannot be moved until the doors are closed. If it were not for switch C' it would be possible for a person at one end of the landings to move the car if he pressed the button during the short interval of time between the stopping of the car and the opening of the landing door. The opening of the door would stop the car, but by this time it might be a foot or two away from the floor level. The current that passes from switch C' to wire r is kept down to a small amount by passing it through a high resistance which in the diagram is marked 700 *w*.

The electrical portion of the Otis electric elevators has been supplied for many years to four or five of the leading companies, which were controlled by the Otis, and during the last two or three years it has been supplied to practically all the prominent makers, as these are now part and parcel of this company; hence the descriptions given in the foregoing are more than likely to cover any case met with in practice, for although there are numerous small manufacturers, the sum total of their elevators in use is comparatively small. The only electric elevators in addition to those described in the foregoing that have come into extensive use are those made by the Sprague Electric Co.

These machines are of two different types, one being the ordinary drum design, and the other the screw machine. The drum machine is similar in its main features to the same type of machine of other makers, and it is only in the minor details of construction that any radical difference can be noted. In the means employed for controlling the motion of the motor, however, there is a decided difference. In all the Sprague elevators the car is controlled electrically, hand rope control not being used in any

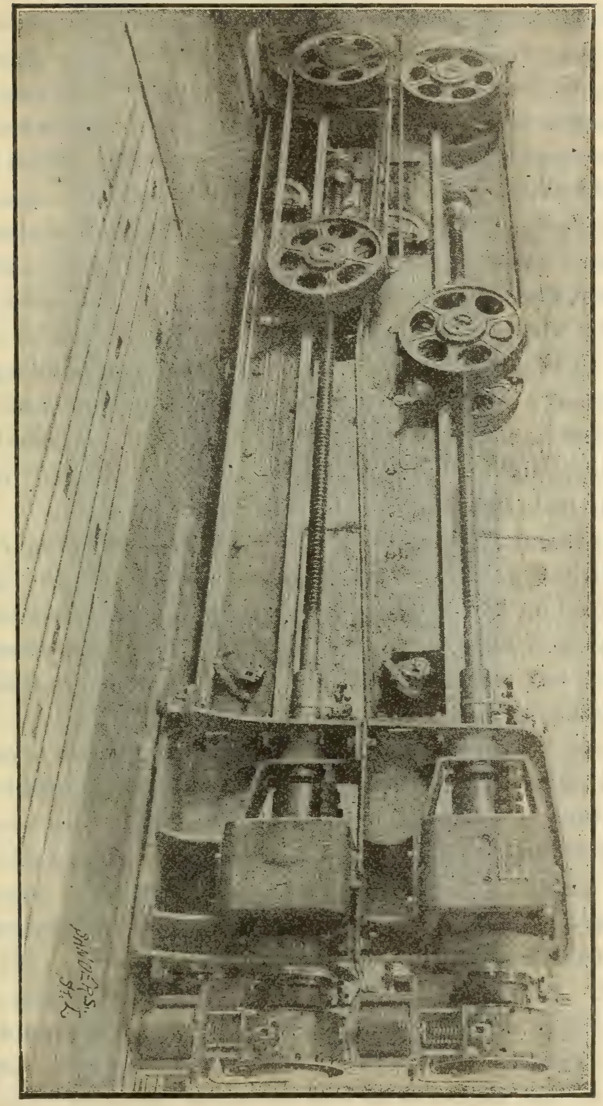


Fig. 334. Sprague screw type electric elevator.

case. The drum machines are arranged like those of other makes, so that the motor is connected with the main line whether the car is going up or down, and acts as a motor or as a generator according to the conditions of the load; that is if the load is lifted, the machine acts as a motor, and if the load is lowered, the machine acts as a generator and holds the car back. With the screw type of machine, the arrangement is different, the motor acting as such in raising the load, but on the descent the motor is disconnected with the main line and acts as a generator, developing a current that circulates in a circuit formed by the motor connecting the wires, and which is entirely independent of the main line. In the drum machine, when the motor acts as a generator in lowering a load, the current it generates is sent back into the main circuit, and at all times the machine is connected with the main line, while with the screw type the motor is only connected with the main line when the load is lifted.

The general appearance of the screw type of Sprague elevator is shown in Fig. 334. This illustration represents two machines, one placed on top of the other. In buildings where there is an abundance of floor space, the machines are all set directly upon the floor, but where floor space is limited, they are stacked two, three and even four high.

As can be clearly seen in Fig. 334, a long screw is coupled to the end of the motor armature shaft. This screw threads through a nut that is mounted in a cross-head that carries a number of sheaves around which the lifting ropes pass. At the extreme end of the machine other sheaves are mounted, these being held in stationary supports. The sheaves carried by the cross-head travel from one end of the machine to the other as the screw is rotated. When they are drawn away from the stationary sheaves the elevator car is raised, and when they move toward the stationary sheaves the elevator is lowered. In this respect the action is just the same as in a horizontal cylinder hydraulic elevator.

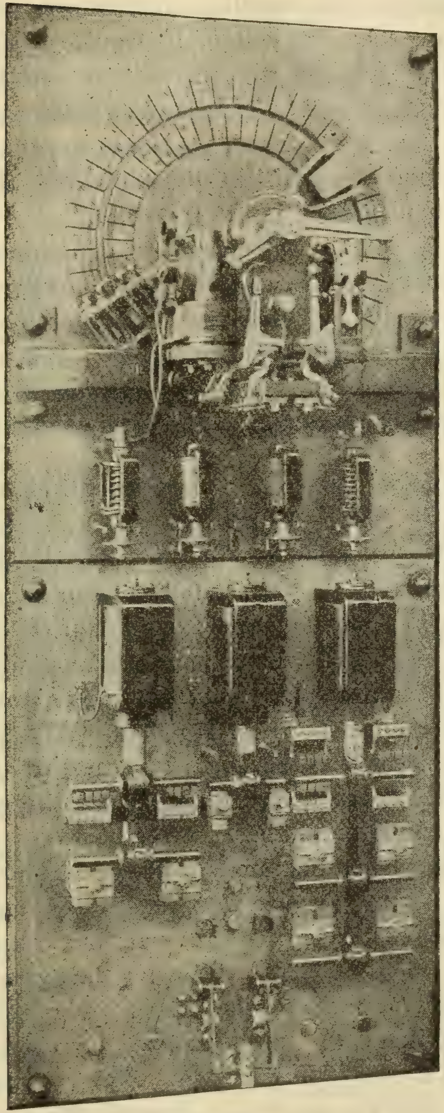


Fig. 335. Elevator controller board.

The nut carried by the traveling cross-head is so arranged that when the latter reaches the end of its travels at either end of the screw, the nut is released and then rotates with the screw without moving the cross-head. This forms a perfect top and bottom limit stop, for even if the motor continues to run, the car cannot be carried beyond the positions corresponding to the points at which the nut slips around in the cross-head.

The brake for holding the machine is mounted upon the outer end of the armature shaft, and can be seen at Fig. 334 at the extreme right hand side. This brake is actuated by a magnet that releases it, and a spring that throws it on. When the current is on, the brake is lifted and when the current is off the brake goes on. In this respect, the action is the same as in all other electric elevators.

The operation of the motor is controlled by a small switch in the car, which is connected with the motor circuits by means of wires contained in a flexible cable, just like the Otis electrically controlled machines. The controller consists of a main switch, which is moved by a small motor called a pilot motor, and a number of smaller magnetic switches whose action will be presently explained. All these parts are mounted upon a switchboard, and present the appearance shown in Fig. 335. The pilot motor and main switch are located at the top of the board, and the magnet switches cover the space below, while the starting and regulating resistance is mounted on the back of the board.

The complete wiring diagrams for these machines is decidedly complicated owing to the fact that there are numerous switches and devices whose office is to afford additional safety, or to render the control more perfect. When all the parts that are not actually necessary to illustrate the system are removed, however, the diagram becomes quite simple and can be readily understood. Such a diagram is shown in Fig. 336. This diagram shows the motor together with the screw and sheaves, the elevator car, the

counterbalance, and the operating switches. The wires marked + and — are connected with the main line. The switch in the car is connected with the controller by means of four wires, marked *c b d* and *s*. The lower one of these wires, marked *s*, is connected with the stud around which the car switch swings. When the car switch is moved onto the upper contact, it connects wire *s* with wire *c* and then the car runs up. When the car switch is moved down onto the lower contact, wire *s* is connected with wire *d*, and then the car runs down. When the car switch is placed in the central position wire *s* is connected with wire *b* and then the elevator stops. The two switches marked "up limit," "down limit," are for stopping the car automatically at the top and bottom landings. Normally the up limit switch is closed and the down limit switch is open. With these switches in this position, which is the position in which they are shown in the diagram, the current from the + wire can pass through the up limit switch to wire *k*, and thence through wire *l* to the armature of the motor, and then through the field coils, and reach wire *m*. It cannot go beyond this point until the switch *C* is moved. This is the main operating switch, which in Fig. 335 is seen at the top of the board, the contacts being arranged in two circles. The pilot motor that rotates the arm of this switch, which is clearly shown in Fig. 335, is represented in this diagram, Fig. 336, at *A*. As will be seen in this diagram, this motor has a field provided with two magnetizing coils, one for the up motion, and one for the down motion, and in addition it is provided with a brake to stop it quickly and hold it when not in use. The portion of the diagram marked *B* is the reversing switch.

Let us suppose now that the car switch is moved upward, so as to cause the elevator to ascend, then wire *s* will be connected with wire *c*. From the + wire a current will pass through wire *a* to *s* and thus to *c*, and through magnet *e* of switch *g*, thus closing this switch so as to connect wires *h* and *i*. The current in wire *c* will

pass to B and through the connecting plate u will reach the end of the up field coil of the pilot motor, and then pass through the armature of this motor, and finally through the magnet that releases the brake. The pilot motor will now rotate the reversing switch B so that the contact plates will move toward the left. This movement will bring plate w under the ends of wires s and i, thus permitting a current from s to pass to i, and as switch g is

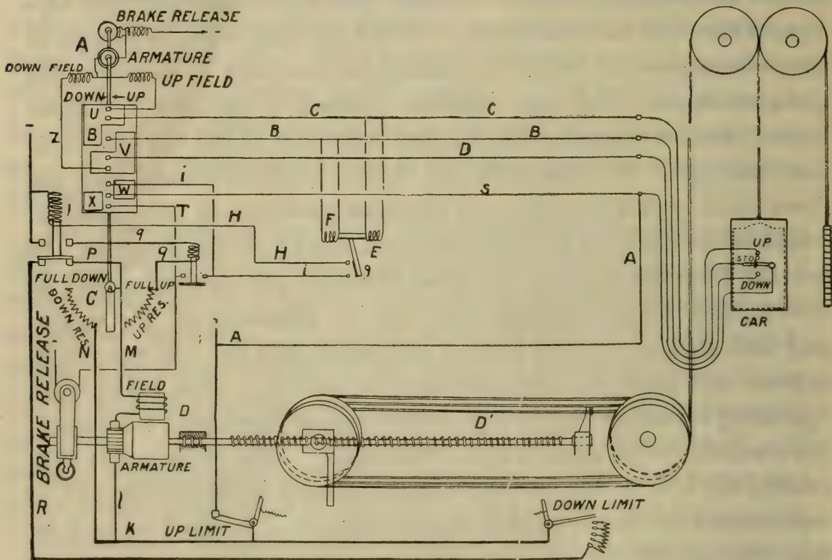


Fig. 336. Diagram of elevator controller board.

closed this current will reach wire h and thus the magnet j, thereby lifting the plunger switch that closes the gap between wire q and the — wire. As the arm of the main switch C moves with the reversing switch B, this arm will ride over the contacts on the right side, marked "up res." and thus the current from wire m will be able to reach wire q after passing through the up resistance.

If the car switch is left on the upper contact, the pilot motor will continue to rotate until the arm of switch C reaches the top of the resistance contacts, marked Full Up. When this point is reached, the contact plate u of the reversing switch B will pass from under wire c and the terminal of the up field of the pilot motor, and then this motor will stop rotating.

If the car switch is not kept on the upper contact very long, the pilot motor can be stopped with the arm of switch C at some intermediate point on the resistance contacts, thus by the time during which the car switch is kept upon the upper contact, the amount of resistance cut out of the motor circuit can be controlled and thereby the speed of the car can be controlled.

In this operation it will be noticed that the motor is connected with the main line and that the current enters through the + wire and passes out through the — wire. If now we turn the car switch downward, the s wire will be connected with the d wire and by following the latter to the reversing switch B it will be seen that through connecting plate v it is connected with wire z which leads to the end of the down field of the pilot motor, thus setting the latter in motion in the opposite direction so as to shift the contact plates of B toward the right, and at the same time rotate the arm of the main switch C to the left, thereby making contact with the contacts of the down resistance. With the arm of C in this position, it will be seen that the current in wire l can flow through the motor armature and field and through wire m to the arm of switch C and through the down resistance to wire n and thus back to wire l, thereby forming a closed circuit within the motor wires and connections, and disconnected from the main line except on the side of the + wire. The rotation of B causes the connecting plate x to ride upon the terminals of wires s and t, and thus a current is sent through the brake magnet so as to lift the brake, and allow the elevator machine to run. When the pilot motor moves the arm of C so far as to reach the top of the down

resistance, the contact plate v of the reversing switch B will pass beyond the terminals of wires d and z, thus breaking the circuit of the pilot motor and bringing the latter to a stop.

When the reversing switch B is in the stop position, as shown in the diagram, the terminal of wire b does not rest upon a connecting plate but when the switch is rotated for the up motion, the terminal of b rests on plate v so that if the car switch is turned to the stop position, the current from wire b will pass to wire z and thus reverse the direction of rotation of the pilot motor, and return the switches to the stop position. If the car is running down, when the car switch is turned to the stop position, the current from wire b will pass to wire z and thus reverse the direction of rotation of the pilot motor, and return the switches to the stop position. If the car is running down, when the car switch is turned to the stop position, the wire b will ride over the plate u and thus the current will pass through the pilot motor through the up field and thus rotate the switches back to the stop position. In each case, as will be noticed, whenever the current flows through wire b it energizes coil f and thus opens switch g. When the car is running up the current for the brake magnet passes from wire i through the switch which is energized by the main current flowing in wire q. When the car runs too far down, and closes the down limit switch, the motor circuit becomes closed through wires p, r and k, thus giving another path for the current generated by the motor armature and thereby increasing the resistance to rotation.

The controller for the Sprague drum machines is very similar to the one just described. It is operated by a pilot motor, and in so far as the controller switchboard is concerned looks the same. The only difference is that rendered necessary by the fact that in lowering as well as in raising the load, the motor is connected with the line. This requires a slight change in some of the wire connections.

The electrical parts of the Sprague elevators require very little attention other than to keep them clean and all the contacts bright and in proper adjustment, so that when moved a good contact may be made. Of the mechanical portion, the drum machines require about the same attention as other machines of this type. As to the screw machines, the part that requires most attention is the screw and the nut. As can be readily understood, if the nut were solid, the friction against the screw would be very great; therefore, to reduce this friction, the nut is made so as to carry a large number of friction balls. These run in a groove cut in the side of a thread and roll between the thread and the screw and the thread in the nut. A tube is attached to the nut to provide a path through which the friction balls can pass from the end of the thread to the beginning, thus making an endless path in which they move. As these friction balls are subjected to a heavy pressure, there is more or less danger of their giving trouble and on that account the thread on the screw should be carefully examined and kept as clean and free from grit as possible. Under favorable conditions these screws run very well, the wear being trifling, but in some instances they are liable to cut badly, hence they should be closely watched.

DIRECTIONS FOR THE CARE AND OPERATION OF THE ELECTRIC ELEVATOR.

Whenever the attendant wishes to handle the machine to clean, adjust, repair or oil it, he should see that the current is shut off at the switch, and thus prevent all possibility of accident.

Cleaning. — Keep the entire machine clean. Clean the commutator and other contacts and brushes carefully with a clean cloth and keep them free from grease and dirt. If the face of the rheostat on which the rheostat arm brushes work becomes burnt, clean with a piece of fine sand-paper (No. 0), or if necessary use

a fine file. Keep all contacts smooth. Try the rheostat arm when cleaning to be sure that it moves freely off contacts.

Oiling. — Oil the drum shaft bearings with good heavy oil. Oil the worm and gear by filling the chamber around them with a mixture of two parts of good castor oil and one part good cylinder oil. Keep this chamber filled to the top of worm or mark on gauge glass, adding a little each day as it is used. The end thrust bearings of the machine are automatically oiled from this chamber. This should be drawn off every two or three months and replaced by fresh oil. Oil the motor bearings with dynamo oil. These are automatically oiled, but should occasionally be supplied with fresh oil. Lubricate the commutator, rheostat face, drum switch and contacts VERY SPARINGLY with a cloth moistened with oil. Care should be taken not to supply too much oil to these parts. Keep the oil dash-pot, if any, sufficiently filled with oil to allow the rheostat arm to move quickly on to the first contact and to retard this movement beyond this contact. The best oil for this purpose is fish oil, or some thin oil that is not readily affected by changes in temperature. If an air dash-pot is used, keep it slightly oiled so as to keep the packing soft. Keep all parts of the elevator, including sheaves, guides, cables, etc., clean and well oiled.

Operating. — Before switching the current on to the machine, be sure that the operating lever is in its central position. To ascend, draw the lever the full throw to the *up*. To descend, draw the lever the full throw to the *down*. To run at slow speed, bring the lever toward the center according to the speed desired. To stop, bring lever to slow speed when within four feet of landing, and to its central position when close to it. In this way, the operator can make accurate stops. When starting machines on which the solenoid is used if the current is admitted to the motor too rapidly, thereby starting the car with a jerk, or momentarily dimming the lights on the circuit, check the speed with which the

resistance is cut out of the armature circuit by slightly easing off the weight, which acts in opposition to the core of the small solenoid. This solenoid controls a valve in the dash-pot and thereby regulates its speed in proportion to the current passing. If a governor starter is used and the current is admitted too rapidly, tighten the governor spring on the armature shaft, or close the vent in air dash-pot. If the car refuses to ascend with a heavy load, immediately throw the lever to the center and reduce the load, as in all probability it is greater than the capacity of the elevator. If it refuses to ascend with a light load, throw the lever to the center and have the fusible strip examined. If, in descending, the car should stop, throw the lever to the center and examine safeties, fusible strip and machine, and before starting, be sure that the cables have not jumped from their right grooves. If the car refuses to move in either direction, throw the lever on the center and have the fusible strips examined. Never leave the car without throwing the lever to the center. If the car should be stalled between floors, it can be either raised or lowered by raising the brake and running it by turning the brake-wheel by hand. Such a stoppage might be caused by the current being shut off at the station, undue friction in the machine, too heavy a load, fuses burnt out, or a bad contact of the switches, binding posts or electrical connections. If the car by any derangement of cables or switch cannot be stopped, let it make its full trip, as the automatic stop will take care of it at either end of the travel. The bearings should be examined occasionally to insure no heating and proper lubrication.

General directions. — Have the machine examined occasionally by someone well posted in electric motors and elevators. The attendant should inspect the machine often. All brushes and switches should be sufficiently tight to give a good contact, but no tighter. None of the brushes should spark when in their

normal position. When the brushes become burnt dress with sandpaper or file, or, if necessary, replace with new ones. If brushes spark, dress with sandpaper or file to a good bearing, and, if necessary, set up springs, but do not make the tension such as to interfere with their ready movement. Adjust commutator brushes gradually for least sparking. These should be close to the central position. Contacts and brushes should be kept clean and smooth and lubricated sparingly. While replacing a fusible strip, be sure that main switch is open, and be careful not to touch the other wire with your tool or otherwise, as such contact would be dangerous. Never put in a larger fuse than the one burnt. Inspect the worm and worm-wheel occasionally through hand-holes in casing, to see that they are well lubricated, and that no grit gets into the oil. They should show no wear. The stuffing-box on the worm shaft should be only tight enough to keep the oil from leaking out of the worm chamber. Be sure that all parts are properly lubricated, and that none of the bearings heat. To make sure that the car and machinery run freely, lift brake lever and then rotate worm shaft by pulling on the brake wheel. The empty car should ascend without any exertion. Keep operating cables properly adjusted. Open main switch when the elevator is not in service.

The Lever.

Relative Position of Power, Weight and Fulcrum in:

Lever of the first class	Power.	Fulcrum.	Weight.
Lever of the second class	Power.	Weight.	Fulcrum.
Lever of the third class	Weight.	Power.	Fulcrum.

$$\frac{\text{Power} \times \text{power-arm}}{\text{Weight-arm}} = \text{Weight.} \qquad \frac{\text{Power} \times \text{power-arm}}{\text{Weight}} = \text{Weight-arm.}$$

$$\frac{\text{Weight} \times \text{weight-arm}}{\text{Power-arm}} = \text{Power.} \qquad \frac{\text{Weight} \times \text{weight-arm}}{\text{Power}} = \text{Power-arm.}$$

$$\frac{\text{Weight}}{\text{Power}} = \text{ratio of, or proportion of, power-arm to weight-arm.}$$

$$\frac{\text{Power-arm}}{\text{Weight-arm}} = \text{ratio of, or proportion of, weight to power.}$$

CHAPTER XXVI.

HYDRAULIC ELEVATORS.

The purpose of these pages is to furnish such instructions and information as will be of use to engineers in the handling of elevator machinery. To accomplish this end, cuts and sectional views of cylinders and valves of the different types of elevator machinery made by the different elevator companies, are herein produced,

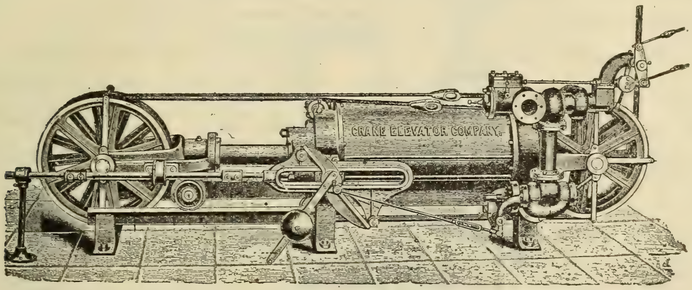


Fig. 337. Horizontal hydraulic passenger machine.

so as to make the different elevators plain to the engineer. It must be borne in mind that the one point of paramount importance for the successful operation of an elevator is proper care and management; a lack of thorough knowledge of the machine and lack of attention in this respect shortens the life of the machine and often makes extensive repairs necessary.

HOW TO PACK HYDRAULIC VERTICAL CYLINDER ELEVATORS.

Packing vertical cylinder piston from top. — Run the car to the bottom and close the gate valve in the supply pipe. Open the air cock at the head of the cylinder, and also keep open the

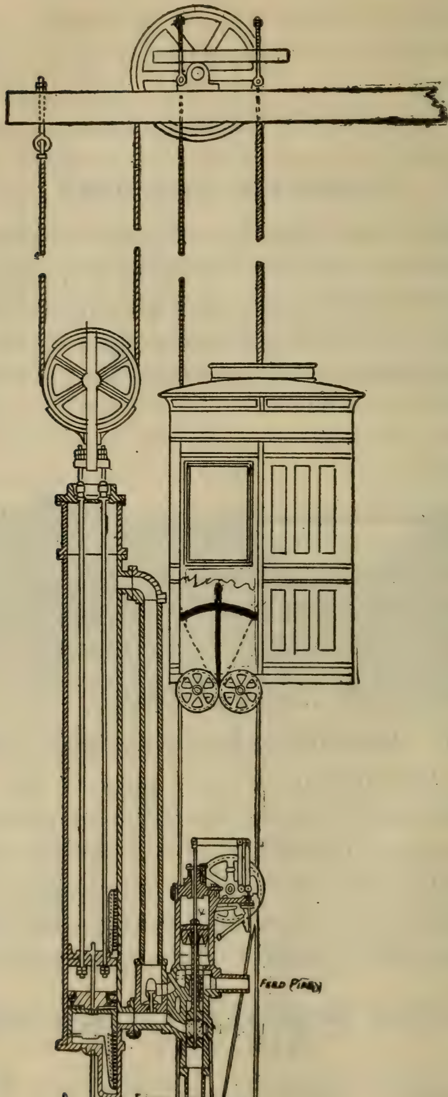


Fig. 338. How to set the rope and hand lever on elevator car; the sheaves should be at the center of the travel, as shown.

valve in the drain pipe from the side of the cylinder long enough to drain the water in the cylinder down to the level of the top of the piston. Now remove the top head of the cylinder, slipping it and the piston rods up out of the way, and fasten there. If the piston is not near enough to the top of the cylinder to be accessible, attach a rope or small tackle to the main cables (not the counter-balance cables) a few feet above the car, and draw them down sufficiently to bring the piston within reach. Remove the bolts in the piston follower by means of the socket wrench furnished for that purpose. Mark the exact position of the piston follower before removing it, so that there will be no difficulty in replacing it. On removing the piston follower we will find a leather cup turned upwards, with coils of $\frac{5}{8}$-inch square duck packing on the outside. This we will remove and clean out the dirt; also clean out the holes in the piston through which the water acts upon the cup. If the leather cup is in good condition, replace it, and on the outside place three new coils of $\frac{5}{8}$-inch square duck packing, being careful that they break joints, and also that the thickness of the three coils up and down does not fill the space by $\frac{1}{4}$ inch, as in such case the water might swell the packing sufficiently to cramp it in this space, thus destroying its power to expand. If too tight, strip off a few thicknesses of canvas. Replace the piston follower and let the piston down to its right position. Replace the cylinder head and gradually open the gate valve in the supply pipe, first being sure that the operating valve is on the down stroke or it is so the car is coming down. As soon as the air has escaped before closing the air cock to make sure the air is all out of the cylinder, make a few trips, and the elevator is ready to run.

Packing the vertical cylinder valves. — To pack the valve, run the car to the bottom and close the gate valve in the supply pipe. Then throw the operating valve for the car to go up, open the air cock at the head of the cylinder and the valve in the drain pipe at the bottom, and the water will drain out of the cylinder.

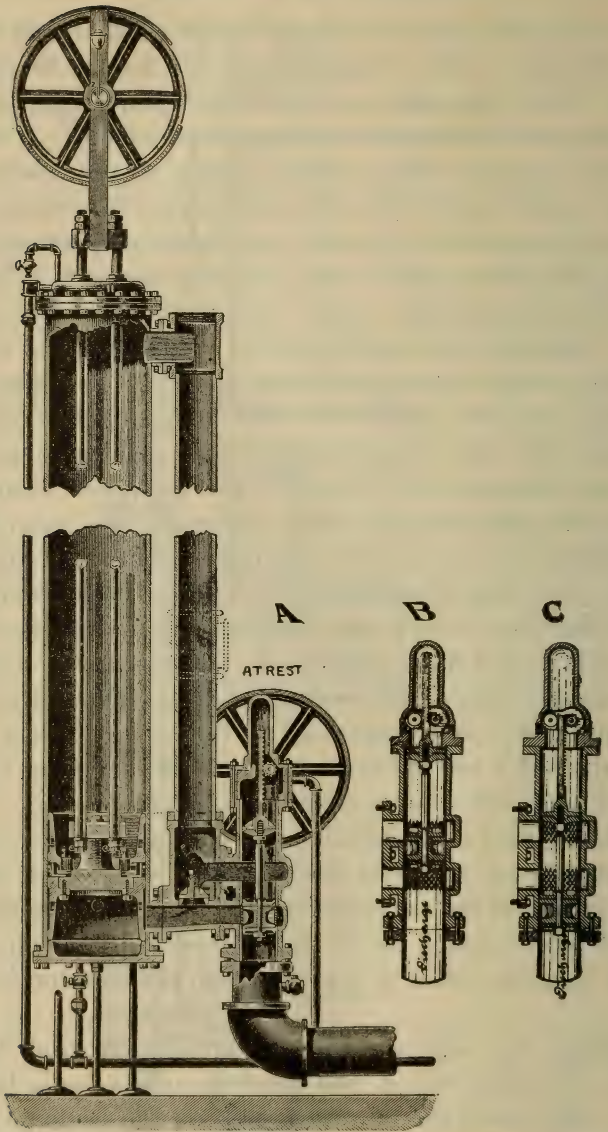


Fig. 339. Otis vertical hydraulic passenger and freight machine.

A shows the position of the valve at rest. *B* shows the position of the valve when the car is going up or hoisting. *C* shows the position of the valve when the car is coming down or lowering.

When the cylinder is empty, reverse the valve for the car to run down, so as to let the water out of the circulating pipe. In cases of tank pressure, where the level of the water in the lower tank is above the bottom of the cylinder, the gate valve in the discharge pipe will have to be closed as soon as the water in the cylinder is on a level with that in the tank, allowing the rest to pass through the drain pipe to the sewer. As soon as the water has all drained off, take off the valve cap and remove the pinion shaft and sheave, marking the position of the sheave and the relation which the teeth on the pinion bear to the teeth on the rack before removing. We can now take out the valve plunger and put the new cup leather packings on in the same position we found the old ones. Replace all the parts as first found. Before refilling the cylinder, close the valves in the drain pipes, but leave the air cock at the head of the cylinder open and be careful that the operating valve is in position for the car to go down. Gradually open the gate valve in the supply pipe. When the cylinder has filled with water and the air has escaped, close the air cock and open the gate valve in the discharge pipe.

Packing piston rods. — Close the gate valve in the supply pipe. Remove the followers and glands to the stuffing-boxes and clean out the old packing. Repack with about eight turns of $\frac{1}{4}$ inch flax packing to each rod, and replace glands and followers. Screw down the followers only tight enough to prevent leaking.

Packing Otis vertical piston from bottom. — Remove the top stop-button on hand rope and run the car up until the piston strikes the bottom head in cylinder. Secure the car in this position by passing a strong rope under the girdle or crosshead and over the sheave timbers. When secured, close the gate valve in the supply pipe, open the air cock at the head of the cylinder, and throw the operating valve for the car to go up. Also open the valve in the drain pipe from the side of the cylinder, and from the lower head of the cylinder, thus allowing the water to drain

out of the cylinder. When the cylinder is empty, throw the valve for the car to descend in order to drain the water from the circulating pipe. In case of tank pressure, where level of water in lower tank is above the bottom of the cylinder, the gate valve in the discharge pipe will have to be closed as soon as the water in the cylinder is on a level with that of the tank, allowing the rest to pass through the drain pipe to the sewer. When the water is all drained off, remove the lower head of the cylinder, and the piston will be accessible. Remove the bolts in the piston follower by means of the socket wrench, which is furnished for that purpose. Before removing the piston head, mark its exact position, then there will be no difficulty in replacing it; also be careful and not let the piston get turned in the cylinder, so as to twist the piston rods. On removing the piston follower, we will find a leather cup turned upwards, with coils of $\frac{5}{8}$ in. square duck packing on the outside. This we will remove and clean out the dirt; also clean out the holes in the piston, through which the water acts upon the cups. If the leather cup is in good condition, replace it and on the outside place three new coils of $\frac{5}{8}$ inch square duck packing, being careful that they break joints and also that the thickness of the three coils up and down does not fill the space by $\frac{1}{4}$ inch, as in such case the water might swell the packing sufficiently to cramp it in this space, thus destroying its power to expand. If too tight, strip off a few thicknesses of canvas. Replace the piston follower and cylinder head, and the cylinder is ready to refill. Close the valves in the drain pipes, leave the air cock open at the head of the cylinder and the operating valve in the position to descend, and open gate valve in the discharge. Slowly open the gate valve in the supply pipe, allowing the cylinder to fill gradually and the air to escape at the head of the cylinder. When the cylinder is full of water, leave the air cock open and put the operating valve on the center. The car can then be untied, the stop button can be reset, and the elevator is ready to use. Make a few trips before closing the air valve.

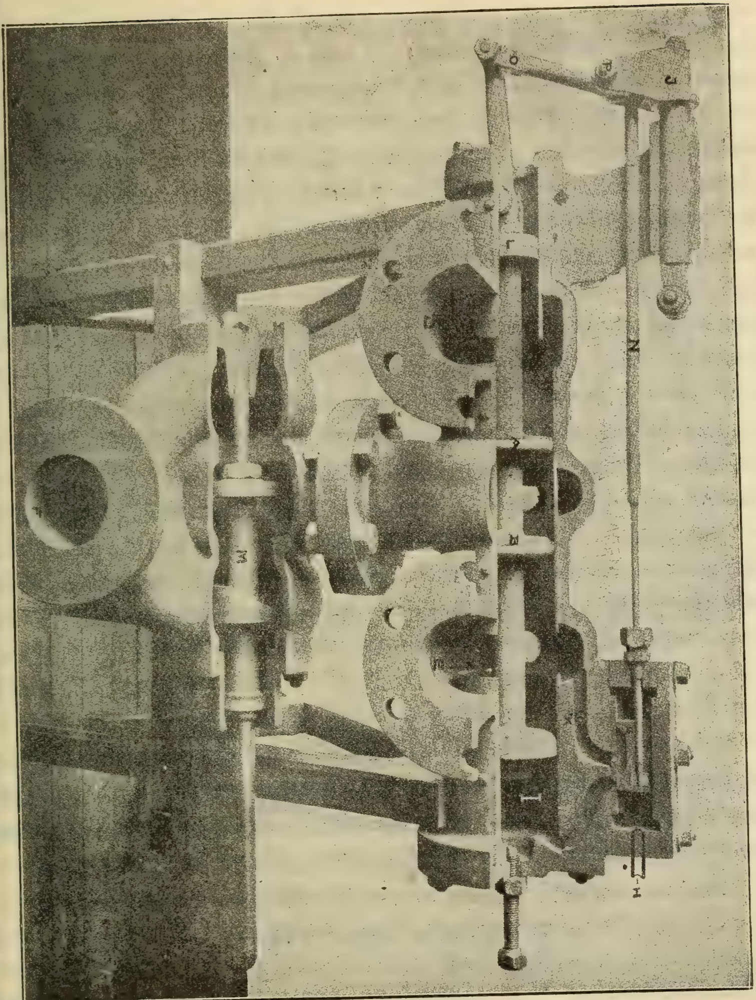


Fig. 340. Auxiliary valve for Crane hydraulic passenger elevators.

The operation of this valve is explained as follows: *D* represents the supply inlet; *E*, the discharge outlet; *F*, the opening

to the cylinder; G, the pilot valve; H, the pilot valve supply pipe to the motor cylinder; N and J, the attachment by which the valve is operated. Fig. 340 represents the valve on centers, or the car at rest at any floor between limits of travel. It will be noticed in cut that the plunger heads A and B are on either side of the central opening. The water is then entirely cut off from the machine and the pilot valve covers the port C. To start the car up, water is admitted to the cylinder I through the inlet D. This is accomplished by pushing on the connection in which opens the port C in the pilot valve G, allowing the water in the motor cylinder I to flow into the discharge E. The flow is regulated by the screw K. The pressure in the motor cylinder I being relieved, the valve plunger moves to the right under the difference in pressure upon the plunger A and L, L being of smaller diameter than A. Supply is thus admitted to the cylinder through F. To start the car down, pull on the connection J. The port C in the pilot valve chest is opened, allowing water from the pilot supply H to flow into the motor cylinder I. The pressure on head forces the plunger B to move to the left. Water is thus allowed to pass out from F to the discharge E. If a slow movement of the car is desired, connection J is removed to the right or left for either up or down, and only enough to open the main valve slightly to give the desired speed. This speed is maintained by the lever O being moved on its fulcrum P, thus necessitating the valve G covering port C.

AUTOMATIC STOP VALVE.

The stop valve M is opened automatically by the machine as the elevator starts from the top or bottom landing, giving free flow of water to the cylinder. As the car reaches the upper or lower limit of travel, the valve is automatically closed, so that the car stops gradually at the terminals.

OTIS GRAVITY WEDGE SAFETY.

Under the car is a heavy hardwood safety plank, on each end of which is an iron adjustable jaw, inclosing the guide on the guide post. In this jaw is an iron wedge, withheld from contact with the guide in regular duty. Under the wedge is a rocker arm, or equalizing bar, with one of the lifting cables attached independently at each extremity. The four lifting cables, after being thus attached, pass over a wrought iron girdle at the top of the car. Each cable carries an equal strain, and the breakage of any one cable puts the load on the other cables, which throws the rocker out of equilibrium and forces the wedges on both sides instantly and immovably between the iron jaws of the safety plank and the side of the guides, stopping the car. It may be raised to any position by the unbroken cables, though it cannot be lowered until a new cable is put on.

Any cable will always stretch before it breaks, which will throw the equalizing safety-bar out of equilibrium and force the wedges on both sides into position. *Few other safety devices will give warning in advance.*

CARE OF HALE ELEVATORS.

Keep the guide springs on the girdle above, and the safety plank below the car adjusted, so that the car will not wabble, but not tight enough to bind against guides. When cables are drawing alike, the equalizing bars on a passenger elevator should be horizontal, and the set screws free from contact with the finger shaft, but adjusted so that one of them will come in contact with the finger shaft when the equalizing bar is tipped a certain amount either way. If the safety wedges should be thrown in, or rattle, when descending, the cause would be from the stretching or breaking of one of the cables, the action of the governor, or from weakness of either the spring on the finger shaft,

safety-wedge or gummy guides. In the first case, if occasioned by the cable stretching, the cable should be examined thoroughly, and if it shows weakness, a new one put on, otherwise, it can be shortened up, as stated above. In the second case, the car had probably attained excessive speed and the governor simply performed its proper function. In the third case, new springs should be put on and the guides kept clean, for it often happens that the guides are so dirty that the springs cannot well prevent the wedges catching. All the safeties should be kept clean and in good order, so that they will quickly respond when called upon to perform their duty. To loosen the wedges when thrown in, throw the valve for the car to ascend. If the wedges are thrown in above the top landing, remove the button on the hand cable and run the car up until the piston strikes the bottom of the cylinder. If this is not sufficient to loosen the wedges, the car will have to be raised by a tackle. Keep all nuts properly tightened.

If traveling or auxiliary sheave bushing is worn so that sheave binds, or the bushing is nearly worn through, turn it half round, and thus obtain a new bearing. If it has been once turned put in a new bushing. See that the piston rods draw alike. If they do not, it can be discerned by trying to turn the rods with the hand, or by a groaning noise in the cylinder. However, this groaning may also be caused by the packing being worn out, in which case the car would not stand stationary. See that all supports remain secure and in good condition.

WATER FOR USE IN HYDRAULIC ELEVATORS.

In hydraulic elevator service little heed is usually given to the quality of water with which the system is operated. Much loss of power by friction and many dollars spent annually in repairs can be avoided by a little thought and action on this subject. In order to prove the truth of this statement, one has only to obtain

two samples of water, one of soft water and the other of what is commonly known as hard water. For example, take rain water as the first sample and water from the well as the second. Now rub your hands briskly together while holding them immersed in one, and then in the other of these samples. You will instantly realize that the quality of water used in elevator service has much to do with the efficiency of the hydraulic machinery. Water from the service pipes of the city water-works always contains more or less sand and other gritty substances, in suspension, and this grit acts much the same on the packing and metal parts of the apparatus as does a sand blast. Some engineers, having realized the evil effects of water in the state that it is generally used, have attempted to remedy the matter by replacing the water which is lost by leakage or evaporation by the addition of the water which is discharged from the steam traps of the plant; and as this has been distilled, it is almost chemically pure — thus the man who uses distilled water in an elevator system instead of the water containing grit, is simply getting out of one difficulty into another.

It is a well-known fact in chemistry that pure water is a solvent for every known substance, and will especially attack iron to a large degree. Whenever it is practicable, the water for elevator use should be passed through a filter to remove grit before being allowed to pass into the surge tank. In many cases, however, it would be difficult for the engineer to convince the owner of the advisability of buying and installing a filter for this purpose. A simple and somewhat inexpensive remedy is within reach of all — the plentiful use of soap will obviate many of the evil effects of hardness of the water, will double the life of the packing, will reduce the loss by friction, and will, to a large extent, prevent the chattering of the pistons, making the elevators run much smoother. In laboratory practice, the degree of hardness or softness of water is determined by the amount of pure

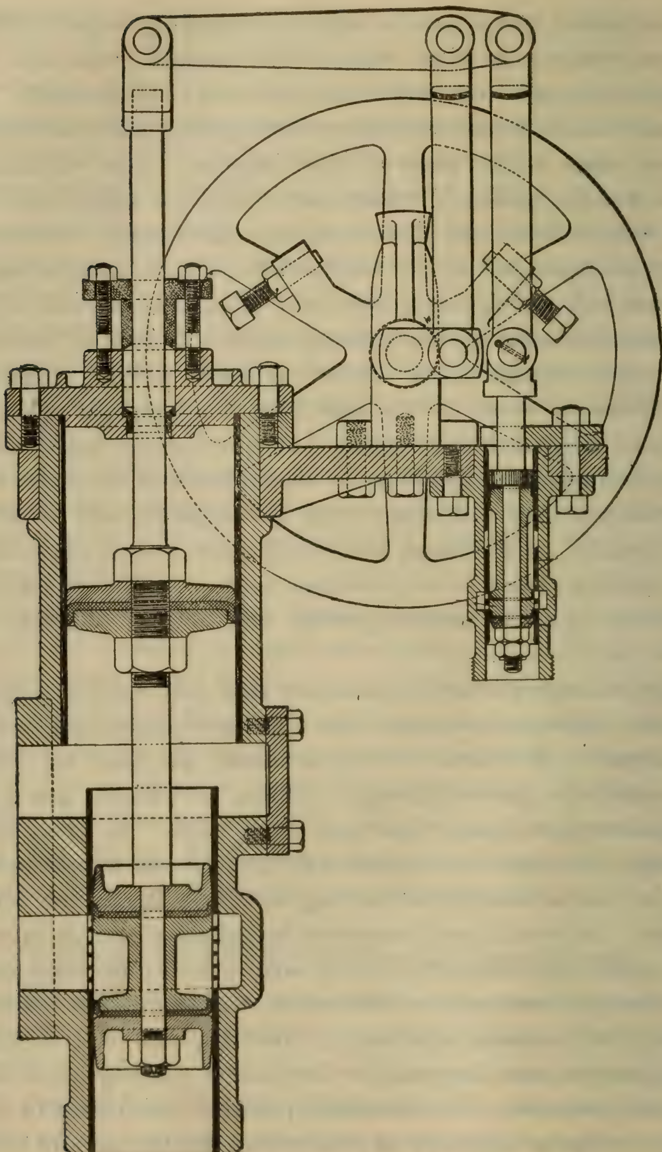


Fig. 341. Otis differential and auxiliary valve.

soap that is necessary to mix with the water to form a lather, or to precipitate a certain quantity of carbonate of lime and other substances. This same action, on a larger scale, takes place when soap is introduced into an elevator tank, and while the oily portion of the soap forms an emulsion with the water, of great lubricating properties, the gritty matter is precipitated and can be gotten rid of through means of a blow-off in the bottom of the tank. The cheapest and most convenient form in which to obtain soap for this purpose, is the soap powder extensively manufactured by various firms and which can be purchased for about four cents per pound. In a plant of six elevators, with usually a storage capacity of some 8,000 gallons, it is a good practice to use about twenty pounds of this soap each week. The soap should be at first dissolved in about ten times its weight of boiling water, and when cold it will form a stiff soft soap. The practice of putting in the refuse oil collected from the drip pans is of little value; it will not mix with the water, but floats on the surface. It rarely gets low enough to enter the suction pipes of the pumps, and has little or no tendency to precipitate the solid matter that is held in suspension in the water.

If car settles, the most probable cause is that the valve or piston needs repacking. If packing is all right, then the air valve in the piston does not properly seat. If the car springs up and down when stopping, there is air in the cylinder. When there is not much air, it can often be let out by opening the air cock and running a few trips, but when there is considerable air, run the car to near the bottom, placing a block underneath for it to rest upon, then place the valve for the car to descend. While in this position, open the air cock and allow the air to escape. This may have to be repeated several times before the air is all removed.

Keep the cylinder and connections protected from frost. Where exposed, the easiest way to protect the cylinder is by an

air-tight box, open at the bottom, at which point keep a gas jet burning during cold weather. Where there is steam in the building, run a coil near the cylinder. Keep stop buttons on hand cable properly adjusted, so that the car will stop at a few inches beyond either landing, before the piston strikes the head of the cylinder. Regulate the speed desired for the car by adjusting the back stop buttons, so that the valve can only be opened either way sufficiently to give this speed. Occasionally try the governor to see that it works properly. Keep the machinery clean and in good order.

ELEVATOR INCLOSURES AND THEIR CARE.

Elevator inclosures, while intended for protection to passengers, are often carelessly neglected and are often a source of danger, unless looked after and taken care of in a proper manner. It is of the utmost importance that no projection of any kind shall extend into the doorways for clothing of passengers to catch on, thus endangering their lives. The door should move freely to insure their action at the touch of the operator. See that all bolts and screws are tight, and replace at once all that fall out, otherwise, the doors and panels may swing into the path of the elevator cage and be torn off, and probably injure some one, thus placing the owner liable to damages. Elevator doors that are automatic in their closing are the best, but all operators should be held strictly responsible for accidents occurring from the carelessness of leaving doors open. All inclosures should be equipped with aprons above the doors to the ceiling and as close to the cage as possible, to prevent passengers from falling out or extending their person through to be caught by ceilings or beams in the elevator shaft. As a rule, proprietors of buildings take a pride in keeping their inclosures and cars in a neat condition, as they are considered an ornament to the building for the purpose for which they are intended, and no expense is spared in the

line of art; so it is recommended that they be kept free from dampness. Dust with a feather duster and use soft rags for cleaning. Never use any gritty substance, soaps or oils. If they become damaged, have the maker repair and relacquer them.

STANDARD HOISTING ROPE WITH 19 WIRES TO THE STRAND.

IRON.

Table No.	Diameter.	Circumference in inches.	Weight per foot in lbs. of rope with hemp center.	Breaking strain in tons of 2000 lbs.	Proper working load in tons of 2000 lbs.	Circumference of new Manilla rope of equal strength.	Minimum size of drum or sheave in feet.
1	$2\frac{1}{4}$	$6\frac{3}{4}$	8.00	74	15	14	13
2	2	6	6.30	65	13	13	12
3	$1\frac{3}{4}$	$5\frac{1}{2}$	5.25	54	11	12	10
4	$1\frac{5}{8}$	5	4.10	44	9	11	$8\frac{1}{2}$
5	$1\frac{1}{2}$	$4\frac{3}{4}$	3.65	39	8	10	$7\frac{1}{2}$
$5\frac{1}{2}$	$1\frac{3}{8}$	$4\frac{3}{8}$	3.00	33	$6\frac{1}{2}$	$9\frac{1}{2}$	7
6	$1\frac{1}{4}$	4	2.50	27	$5\frac{1}{2}$	$8\frac{1}{2}$	$6\frac{1}{2}$
7	$1\frac{1}{8}$	$3\frac{1}{2}$	2.00	20	4	$7\frac{1}{2}$	6
8	1	$3\frac{1}{8}$	1.58	16	3	$6\frac{1}{2}$	$5\frac{1}{4}$
9	$\frac{7}{8}$	$2\frac{3}{4}$	1.20	11.50	$2\frac{1}{2}$	$5\frac{1}{2}$	$4\frac{1}{2}$
10	$\frac{3}{4}$	$2\frac{1}{4}$	0.88	8.64	$1\frac{3}{4}$	$4\frac{3}{4}$	4
$10\frac{1}{4}$	$\frac{5}{8}$	2	0.66	5.13	$1\frac{1}{4}$	$3\frac{3}{4}$	$3\frac{1}{2}$
$10\frac{1}{2}$	$\frac{9}{16}$	$1\frac{5}{8}$	0.44	4.27	$\frac{3}{4}$	$3\frac{1}{2}$	$2\frac{3}{4}$
$10\frac{3}{4}$	$\frac{1}{2}$	$1\frac{1}{2}$	0.35	3.48	$\frac{1}{2}$	3	$2\frac{1}{4}$
10a	$\frac{7}{16}$	$1\frac{3}{8}$	0.29	3.00	$\frac{3}{8}$	$2\frac{3}{4}$	2
$10\frac{7}{8}$	$\frac{3}{8}$	$1\frac{1}{4}$	0.26	2.50	$\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{1}{2}$

Operating Cable or Tiller Rope, $\frac{3}{4}$ in. diam.; $\frac{5}{8}$ in. diam.; $\frac{1}{2}$ in. diam.; $\frac{3}{8}$ in. diam.

Cables, and how to care for them. — Wire and hemp ropes of same strength are equally pliable. Experience has demonstrated that the wear of wire cables increases with the speed. Hoisting ropes are manufactured with hemp centers to make them more pliable. Durability is thereby increased where short bending

occurs. All twisting and kinking of wire rope should be avoided. Wire rope should be run off by rolling a coil over the ground like a wheel. In no case should galvanized rope be used for hoisting purposes. The coating of zinc wears off very quickly and corrosion proceeds with great rapidity. Hoisting cables should not be spliced under any circumstances. All fastenings at the ends of rope should be made very carefully, using only the best babbitt. All clevises and clips should fit the rope perfectly. Metal fastenings, where babbitt is used, should be warmed before pouring, to prevent chilling. Examine wire ropes frequently for broken wires. Wire hoisting ropes should be condemned when the wires (not strands) commence cracking. Keep the tension on all cables alike. Adjust with draw-bars and turnbuckles provided.

Leather cup packings for valves. — Leather for cups should be of the best quality, of an even thickness, free from blemish and treated with a water-proof dressing. The cups should be of sufficient stiffness to be self-sustaining when passing over perforated valve lining When ordering cups, the pressure of water carried should be specified, as the stiff cups intended for high-pressure would not set out against the valve lining when low pressure is used.

Water. — Water for use in hydraulic elevators should be perfectly clear and free from sediment. A strainer should be placed on the supply pipe and water changed every three months, and the system washed and flushed.

Closing down elevators. — If an elevator is to be shut down for an indefinite period, run the car to the bottom and drain off the water from all parts of the machine; otherwise, a freeze is likely to burst some part of the machinery. If the machine is of the horizontal type, grease the cylinder with a heavy grease; if vertical, the rods should be greased. Oil cables with raw linseed oil.

LUBRICATION FOR HYDRAULIC ELEVATORS.

The most effectual method of lubricating the internal parts of hydraulic elevator plants where pump and tanks are used, is to carry the exhaust steam drips from the foot of the pump exhaust pipe to the discharge tank, thus saving the distilled water and cylinder oil. This system is invaluable when water holding in solution minerals is used, as these minerals greatly increase corrosion. Horizontal machines operated by city pressure are best lubricated with a heavy grease applied either mechanically or by means of a piece of waste on the end of a pole. The former method serves as a constant lubricator, while in the latter case, greasing is often neglected, and in consequence packing lasts but a short time.

Lubrication of worm gearing. — Oils with a body, such as cylinder and castor oils, are best suited to the purpose. A composition of two parts castor to one part cylinder oil of the very best quality, makes a desirable lubricant, for the following reasons: cylinder oil being heavy with ample body, on becoming warm runs freely to the point of contact between the worm and the gear and lubricates readily. On the other hand, castor oil when cool, or only slightly warm, retains its body and makes an excellent lubricant. Upon becoming heated, castor oil thickens, thus rendering it objectionable. By the combination, efficient lubrication is obtained at all temperatures.

Lubrication of cables. — A good compound for preservation and lubrication of cables is composed of the following: Cylinder oil, graphite, tallow and vegetable tar, heated and thoroughly mixed. Apply with a piece of sheepskin with wool inside. To prevent wire rope from rusting, apply raw linseed oil.

Lubrication of guides. — Steel guides should be greased with good cylinder oil. Grease wood strips with No. 3 Albany grease or lard oil. Clean guides twice a month to prevent gumming.

Lubrication of overhead sheave boxes. — In summer use a heavy grease. In winter, add cylinder oil as required.

CARE OF ELEVATOR BELTS.

The work required of an elevator belt is most severe and we might say extraordinary character, running as it does over a large to a small pulley and beneath an idler, so situated as to give the small pulley as much belt surface as possible. The belt runs forward and backward as the cage descends and ascends, thereby causing a certain amount of slip. It is imperative that a belt performing such service should be of the very best quality. The following are the specifications: The stock should be strictly pure oak-tanned, cut in such a manner that the center of the hide will form the center of the belt. Each piece should have all stretch thoroughly removed. The belt should be short lap, none of the pieces to exceed 4′ 2″ in length, including the laps. Lock lap should be made, which makes a perfect splice. Under no circumstances should a straight lap be used. The cement should be of the very best quality and pliable to such an extent that it will allow for the short turn taken by the belt in passing under the idler and around the small pulley. As a precaution against laps coming apart from accident or other cause, belts should be riveted, as the rivets will hold lap together until defect may be seen and remedied. Owing to the high speed, laced belts should never be used, as the laces are sure to be cut by running over the small pulleys. Castor oil makes a very reliable dressing for belts. It renders them pliable, thus improving the adhesive qualities.

USEFUL INFORMATION.

To find leaks in elevator pressure tanks in which air is confined, paint round the rivet heads with a solution of soap and the leak will be found wherever a bubble or suds appear. To ascertain the number of gallons in cylinders and round tanks, multi-

ply the square of the diameter in inches by the height in inches and the product by .0034 = gallons. Weight of round wrought iron: Multiply the diameter by 4, square the product and divide by 6 = the weight in pounds per foot. To find the weight of a casting from the weight of a pine pattern, multiply the weight of pattern by 16.7, for cast-iron, and by 19 for brass. Ordinary gray iron castings = about 4 cubic inches to the pound.

Water. — A gallon of water (U. S. Standard) contains 231 cu. in. and weighs $8\frac{1}{3}$ lbs. A cubic foot of water contains $7\frac{1}{2}$ gal. or 1728 cu. in. and weighs 62.425 lbs. A "Miner's inch" is a measure for the flow of water and is the amount discharged through an opening 1 inch square in a plank 2 in. in thickness, under a head of 6 in. to the upper edge of the opening; and this is equal to 11.625 U. S. gal. per minute. The height of a column of fresh water, equal to a pressure of 1 lb. per sq. in., is 2.304 feet. A column of water 1 ft. high exerts a pressure of .433 lbs. per sq. in. The capacity of a cylinder in gallons is equal to the length in inches multiplied by the area in inches, divided by 231 (the cubical contents of one U. S. gal. in inches). The velocity in feet per minute, necessary to discharge a given volume of water in a given time, is found by multiplying the number of cu. ft. of water by 144 and dividing the product by the area of the pipe in inches.

Decimal Equivalents of an Inch.

1-16	1-8	3-16	1-4	5-16	3-8	7-16	1-2
.0625	.125	.1875	.25	.3125	.375	.4375	.5
9-16	5-8	11-16	3-4	13-16	7-8	15-16	
.5625	.625	.6875	.75	.8125	.875	.9375	

CHAPTER XXVII.

THE DRIVING POWER OF BELTS.

The average strain or tension at which belting should be run, is claimed to be 55 pounds for every inch in width of a single belt, and the estimated grip is one-half pound for every square inch of contact with pulley, when touching one-half of the circumference of the pulley. For instance a belt running around a 36-inch pulley would come in contact with one-half its circumference, or $56\frac{1}{2}$ inches, and allowing a half-pound per inch, would have a grip $28\frac{1}{4}$ pounds for each inch of width of belt.

MECHANICAL PROBLEMS AND RULES.

Problem 1. To find the circumference of a circle or a pulley: —

Solution. Multiply the diameter by 3.1416; or, as 7 is to 22 so is the diameter to the circumference.

Problem 2. To compute the diameter of a circle or pulley: —

Solution. Divide the circumference by 3.1416; or multiply the circumference by .3183; or as 22 is to 7, so is the circumference to the diameter, equally applicable to a train of pulleys, the given elements being the diameter and the circumference.

Problem 3. To find the number of revolutions of driven pulley, the revolution of driver, and diameter of driver and driven being given: —

Solution. Multiply the revolutions of driver by its diameter, and divide the product by the diameter of driven.

Problem 4. To compute the diameter of driven pulley for any desired number of revolutions, the size and velocity of driver being known: —

Solution. Multiply the velocity of driver by its diameter and divide the product by the number of revolutions it is desired the driven shall make.

Problem 5. To ascertain diameter of driving pulley: —

Solution. Multiply the diameter of driven by the number of revolutions it is desired it shall make, and divide the product by the number of revolutions of the driver.

6. **Rule for finding length of belt wanted:** Add the diameters of the two pulleys together, divide the result by two, and multiply the quotient by 3 1/7. Add the product to twice the distance between the centers of the shafts, and we have the length required.

For Calculating the Number of Horse-Power Which a Belt Will Transmit, Its Velocity and the Number of Square Inches in Contact With the Pulley Being Known.

Divide the number of square inches of belt in contact with the pulley by two, multiply this quotient by velocity of the belt in feet per minute and divide the product by 33,000; the quotient is the number of horse-power.

Example. — A 20-inch belt is being moved with a velocity of 2,000 feet per minute, with six feet of its length in contact with the circumference of a four-foot drum; desired its horse-power. 20 x 72 equal 1,440, divided by two, equals 720, x 2,000 equal 1,440,000, divided by 33,000 equals $43\frac{2}{3}$ horse-power.

Rule for finding width of belt, when speed of belt in feet per minute and horse-power wanted are given: —

For single belts. — Divide the speed of belt by 800. The horse-power wanted divided by this quotient, will give the width of belt required.

Example. — Required the width of single belt to transmit 100 horse-power. Engine pulley 72″ in diameter. Speed of engine, 220 revolutions per minute.

800) 4146 (speed of belt per minute).

5.18)100.00 (horse-power wanted).

19″ width of belt required.

For double belts. — Divide the speed of belt in feet per minute by 560. Divide the horse-power wanted by this quotient for the width of belt required.

Example. — Required the width of double belt to transmit 500 horse-power. Engine pulley 72″ in diameter. Speed of engine, 220 revolutions per minute.

560)4146 (speed of belt per minute).

7.4)500.00 (horse-power wanted).

67½″ width of belt required.

NOTES ON BELTS.

PRINCIPLES GOVERNING BELTS.

Although there is not near as much known in general about the power of transmitting agencies as there should be, still it seems that almost any other method or means is better understood than belts.

One of the chief difficulties in the way of a better knowledge of the belting problem, is the relation that belts and pulleys bear to each other. The general supposition, and one that leads to many errors, is that the larger in diameter a pulley is, the greater its holding capacity — the belt will not slip so easily, is the belief. But it is merely a belief, and has nothing to sustain it, unless it be faith, and faith without work is an uncertain factor. It is very

desirable to impress upon the minds of all interested, the following immutable principles or law: —

1. The adhesion of the belt to the pulley is the same — the arc or number of degrees of contact, aggregate tension or weight being the same — without reference to width of belt or diameter of pulley.

2. A belt will slip just as readily on a pulley four feet in diameter, as it will on a pulley two feet in diameter, provided the conditions of the faces of the pulleys, the arc of contact, the tension, and the number of feet the belt travels per minute are the same in both cases.

3. A belt of a given width and making two thousand, or any other given number of feet per minute, will transmit as much power running on pulleys two feet in diameter as it will on pulleys four feet in diameter, provided the arc of contact, tension and conditions of pulley faces all be the same in both cases.

It must be remembered, in reference to the first rule, that when speaking of tensions, that aggregate tension is never meant unless so specified. A belt six inches wide, with the same tension, or as taut as a belt one inch wide, would have six times the aggregate tension of the one inch belt. Or it would take six times the force to slip the six inch belt as it would the one inch. It is well to induce readers to become practical students and to be able to learn for themselves. Information obtained in that way is far more valuable, and liable to last much longer.

In order that the reader may more fully understand whether or not a large pulley will hold better than a small one, let him provide a short, stout shaft, say three or four feet long and two inches in diameter. To this shaft firmly fasten a pulley, say 12 in. in diameter, or any other size small pulley that may be convenient. The shaft must then be raised a few feet from the floor and firmly fastened, either in vices, or by some other means, so that it will not turn. It would be better, of course, to have

a smooth-faced iron pulley, as such are most generally used. So far as the experiment is concerned, it would make no difference what kind of a pulley was used, provided all the pulleys experimented with be of the same kind, and have the same kind of face finish. When the shaft and pulleys are fixed in place, procure a new leather belt and throw it over the pulley. To one end of the belt attach a weight, equal, say, to forty pounds — or heavier, if desired — for each inch in width of belt used; let the weight rest on the floor. To the other end of the belt attach another weight, and keep adding to it until the belt slips and raises the first weight from the floor. After the experimenter is satisfied with playing with the 12 in. pulley, he can take it off the shaft and put on a 24 in., a 36 in., or any other size he may wish; or, what is better, he can have all on the shaft at the same time. The belt can then be thrown over the large pulley and the experiment repeated. It will then be found if pulley faces are alike, that the weight which slipped the belt on the small pulley will also slip it on the large one. The method shows the adhesion of a belt with 180 degrees contact, but as the contact varies greatly in practice, it is well enough to understand what may be accomplished with other arcs of contact. But, after all, many are probably at a loss how to account for some observations previously made. They have noticed that when a belt at actual work slipped, an increase in the size (diameter) of the pulleys remedied the difficulty and prevented the slipping.

A belt has been known to refuse to do the work allotted to it, and continue to slip over pulleys two feet in diameter, but from the moment pulleys were changed to three feet in diameter there was no further trouble. These observed facts seem to be at variance with and to contradict the results of the experiments that have been made. All, however, may rest assured that it is only apparent, not real.

The resistance to slippage is simply a unit of useful effect (or

that which can be converted into useful effect). The magnitude of the unit is in proportion to the tension of the belt. The sum total of useful effect depends upon the number of times the unit is multiplied. A belt 6 inches wide and having a tension equal to 40 lbs. per inch in width, and traveling at the rate of 1 foot per minute, will raise a weight of 240 lbs. 1 foot high per minute. If the speed of the belt be increased to 136.5 feet per minute, it will raise a weight of 33,000 lbs. per minute, or be transmitting 1 horse-power. The unit of power transmitted by a belt is rather more than its tension, but to take it at its measured tension is at all times safe, and 40 to 45 lbs. of a continuous working strain is as much, perhaps, as a single belt should be subjected to. A little reflection will now convince the reader that a belt transmits power in proportion to its lineal speed, without reference to the diameter of the pulleys. Having arrived at that conclusion, it is then easy to understand why it is that a belt working over 36-inch pulley will do its work easy, when it refused to do it and slipped on 24-inch pulleys. If the belt traveled 800 feet per minute on the 24-inch pulleys, on the 36-inch it would travel 1,200 feet, thus giving it one-half more transmitting power. If, in the first instance, it was able to transmit but 8 horse-power, in the second instance it will transmit 12 horse-power. All of which is due to the increase in the speed of the belt and not to the increase in the size of the pulleys; because, as has been shown, the co-efficient of friction, or resistance to slippage, is the same on all pulleys with the same arc of belt contact.

There is no occasion for elaborate and perplexing formulas and intricate rules. They serve no useful purpose, but tend only to mystify and puzzle the brain of all who are not familiar with the higher branches of mathematics, — and it is the fewest number of our every-day practical mechanics who are so familiar. In all, or nearly all treatises on belting, the statement is made that at 600, 800 or 1,000 feet per minute, as the case may be, a belt one

inch wide, will transmit one horse-power; and yet, when we come to apply these rules in practice, no such results can be obtained one time in ten. The rules are just as liable to make the belt travel 400, 1,000 or 1,600 per minute per horse-power as the number of feet they may give as indicating a horse-power.

All of the following simple calculations are based upon the assumption that a belt traveling 800 feet per minute, and running over pulleys, both of which are the same diameters, will easily transmit one horse-power for each inch in width of belt. A belt under such circumstances would have 180 degrees of contact on both pulleys without the interposition of idlers or tighteners.

The last proposition being accepted as true and the basis correct, the whole matter resolves itself into a very simple problem, so far as a belt with 180 degrees contact is concerned. It is simply this: If a belt traveling 800 feet per minute transmit one horse-power, at 1,600 feet, it will transmit two horse-power; or if 2,400 feet, three horse-power, and so on. It is no trouble for any one to understand that, if he understands simple multiplication or division.

It is not, however, always the case that both pulleys are the same size, and as soon as the relative sizes of the pulleys change, the transmitting power of the belt changes; and that is the reason why no general rule has ever, or ever will be made for ascertaining the transmitting capacity of belts under all circumstances. When the pulleys differ in size, the larger of the two is lost sight of — it no longer figures in the calculations — the small pulley, only, must be considered. To get at it, the number of degrees of belt contact on the small pulley must be ascertained as nearly as possible and use for a guide, for getting at the transmitting power, the next established basis given. Of course, the experimenter can make a rule for every degree of variation, but it would require a great many, and is not necessary. Five divisions are used, as follows: —

For 180	degrees	useful	effect	100
For $157\frac{1}{2}$	"	"	"	.92
For 135	"	"	"	.84
For $112\frac{1}{2}$	"	"	"	.76
For 90	"	"	"	.64

The experimenters may find that the figures are under obtained results, which is exactly what they are intended to be, more especially on the 90 degree basis. Always make ample allowance.

To ascertain the power a belt will transmit under the first-named conditions: Divide the speed of the belt in feet per minute by 800, multiply by its width in inches and by 100. For the second, divide by 800, multiply by width in inches and by .92. Third place, divide by 800, multiply by width in inches and by .84. Fourth place, divide by 800, multiply by width in inches and by .76. Fifth place, divide by 800, multiply by width in inches and by .64. As an example: What would be the transmitting power of a 16-inch belt traveling 2,500 feet per minute by each of the above rules?

1st: 2,500	divided by	800	equal	3.125 x 16 x 100	equal 50 h. p.
2d: 2,500	"	800	"	3.125 x 16 x .92	equal 46 "
3d: 2,500	"	800	"	3.125 x 16 x .84	equal 42 "
4th: 2,500	"	800	"	3.125 x 16 x .76	equal 38 "
5th: 2,500	"	800	"	3.125 x 16 x .64	equal 32 "

As stated, when the degrees of contact come between the divisions named above, in order to be on the safe side, calculate from the first rule below it, or make it approximate as desired.

If the above rule is studied well and strictly used, there can be no excuse for any mechanic putting in a belt too small for the work it has to do, provided he knows how much there is to do, which he ought, somewhere near at least.

HORSE-POWER TRANSMITTED BY LEATHER BELTS.

DRIVING POWER OF SINGLE BELTS.

Speed in Feet per Minute.	Width of Belt in Inches.								
	2	3	4	5	6	8	10	12	14
	H. P.	H. P.	H. P.	H. P.	H. P.	H. P.	H. P.	H. P.	H. P.
400	1	1½	2	2½	3	4	5	6	7
600	1½	2¼	3	3¾	4½	6	7½	9	10½
800	2	3	4	5	6	8	10	12	14
1,000	2½	3¾	5	6¼	7½	10	12½	15	17½
1,200	3	4½	7	7½	9	12	15	18	21
1,500	3¾	5¾	7½	9½	11½	15	18¾	22½	26½
1,800	4½	6¾	9	11¼	13½	18	22½	27	31½
2,000	5	7½	10	12½	15	20	25	30	35
2,400	6	9	12	15	18	24	30	36	42
2,800	7	10½	14	17½	21	28	35	42	49
3,000	7½	11¼	15	18¾	22½	30	37½	45	52½
3,500	8¾	13	17½	22	26	35	44	52½	61
4,000	10	15	20	25	30	40	50	60	70
4,500	11¼	17	22½	28	34	45	57	69	78
5,000	12½	19	25	31	37½	50	62½	75	87

DRIVING POWER OF DOUBLE BELTS.

Speed in Feet per Minute.	Width of Belts in Inches.								
	6	8	10	12	14	16	18	20	24
	H. P.	H. P.	H. P.	H. P	H. P.	H. P.	H. P.	H. P	H. P.
400	4¼	5¾	7¼	8½	10	11½	13	14½	17½
600	6½	8¾	11	13	15	17½	19½	22	26
800	8½	11½	14½	17½	20½	23	26	29	34½
1,000	11	14½	18¼	21½	25½	29	32½	36	43½
1,200	13	17½	22	26	30½	34½	39	44	52½
1,500	16¼	21¾	27¼	32½	38	43½	49	54½	65½
1,800	19½	26	32¾	39	45½	52	59	65½	78½
2,000	21¾	29	36½	43½	50½	58	65½	72½	87
2,400	26	34¾	44	52½	60½	69½	78½	88	105
2,800	30½	40½	51	61	71	81	91½	102	122
3,000	32½	43½	54½	65½	76	87½	98	108	131
3,500	38	50¾	63½	76	89	101	114	127	153
4,000	43½	58¼	72¾	87	101	116	131	145	174
4,500	49	65	82	98	114	131	147	163	196
5,000	54½	72¾	91	109	127	145	163	182	218

Example. — Required the width of a single belt, the velocity of which is to be 1,500 feet per minute; it has to transmit 10 horse-power, the diameter of the smaller drum being four feet with five feet of its circumference in contact with the belt.

33,000 x 10 equal 330,000, divided by 1,500 equal 220, divided by 5 equal 44, divided by 6 equal $7\frac{1}{3}$ inches, the required width of belt.

Directions for calculating the number of horse power which a belt will transmit. Divide the number of square inches of belt in contact with the pulley by two; multiply this quotient by the velocity of the belt in feet per minute; again we divide the total by 33,000 and the quotient is the mumber of horse-power.

Explanations. — The early division by two is to obtain the number of pounds raised one foot high per minute, half a pound being allowed to each square inch of belting in contact with the pulley.

Example. — A six-inch single belt is being moved with a velocity of 1,200 feet per minute, with four feet of its length in contact with a three-foot drum. Required the horse-power.

6 x 48 equal 288, divided by 2 equal 144 x 1,200 equal 172,-800, divided by 33,000 equal, say, $5\frac{1}{4}$ horse-power.

It is safe to reckon that a double belt will do half as much work again as a single one.

Hints to users of belts. — 1. Horizontal, inclined and long belts give a much better effect than vertical and short belts.

2. Short belts require to be tighter than long ones. A long belt working horizontally increases the grip by its own weight.

3. If there is too great a distance between the pulleys, the weight of the belt will produce a heavy sag, drawing so hard on the shaft as to cause great friction at the bearings; while, at the same time, the belt will have an unsteady motion, injurious to itself and to the machinery.

4. Care should be taken to let the belts run free and easy, so

as to prevent the tearing out of the lace holes at the lap; it also prevents the rapid wear of the metal bearings.

5. It is asserted that the grain side of a belt put next to the pulley will drive 30 per cent more than the flesh side.

6. To obtain a greater amount of power from the belts the pulleys may be covered with leather; this will allow the belts to run very slack and give 25 per cent more durability.

7. Leather belts should be well protected against water, oil and even steam and other moisture.

8. In putting on a belt, be sure that the joints run with the pulleys, and not against them out.

9. In punching a belt for lacing, it is desirable to use an oval punch, the larger diameter of the punch being parallel with the belt, so as to cut out as little of the effective section of the leather as possible.

10. Begin to lace in the center of the belt and take care to keep the ends exactly in line and to lace both sides with equal tightness. The lacing should not be crossed on the side of the belt that runs next the pulley. Thin but strong laces only should be used.

11. It is desirable to locate the shafting and machinery so that belts shall run off from each other in opposite directions, as this arrangement will relieve the bearings from the friction that would result where the belts all pull one way on the shaft.

12. If possible, the machinery should be so planned that the direction of the belt motion shall be from the top of the driving to the top of the driven pulley.

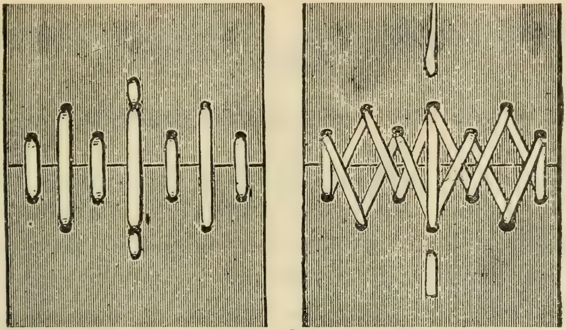
13. Never overload a belt.

14. A careful attention will make a belt last many years which through neglect might not last one.

DIRECTIONS FOR ADJUSTING BELTING.

In lacing, cut the ends perfectly square, else the belt will stretch unevenly. Make two rows of holes in each end; put the

ends together and lace with lace leather, as shown in Figs. 342 and 343. For wide belts, in addition, put a thin piece of leather or



Figs. 342 and 343. Showing laced joint.

rubber on the back to strengthen the joint, equal in length to the width of the belt, and sew or rivet it to the belt. In putting on belting, it should be stretched as tight as possible, and with wide belts, this can be done best by the use of belt clamps.

HORSE POWER OF BELTING.

To ascertain horse-power which belts will transmit, multiply width of belt by diameter of pulley in inches, by revolutions of pulley per minute, by number in table corresponding to the pull the belt can exert per inch of width.

Example. — 10″ single horizontal belt, 36″ pulley, 200 revolutions, pull taken at 50 lbs.

$$10'' \times 36'' \times 200 \times 0.0004 = 28.8 \text{ horse-power.}$$

The pulls which belts 1″ wide will transmit are as follows: —

		Constant.
Single horizontal belts (pulleys same diameter)	50 lbs.	.0004
Double " " " " "	100 "	.0008
Single vertical " " " "	40 "	.00032
Double " " " " "	60 "	.0005
Single belts (large to very small pulleys) .	10 "	.0001
Double " " " " .	15 "	.00012
Quarter twist, single belts	25 "	.0002
" " double "	40 "	.00032

CHAPTER XXVIII.

CAPACITY OF AIR COMPRESSORS.

To ascertain the capacity of an air compressor in cubic feet of free air per minute, the common practice is to multiply the area of the intake cylinder by the feet of piston travel per minute. The free air capacity of the compressor, divided by the number of atmospheres, will give the volume of compressed air per minute. To ascertain the number of atmospheres at any given pressure, add 15 lbs. to the gauge pressure; divide this sum by 15 and the result will be the number of atmospheres. The above method of calculation, however, is only theoretical and these results are never obtained in actual practice, even with compressors of the very best design working under the most favorable conditions obtainable. Allowances should be made for losses of various kinds, the principal losses being due to clearance spaces, but in machines of poor design and construction other losses occur through imperfect cooling, leakages past the piston and through the discharge valves, insufficient area and improper working of inlet valves, etc. In practice there are compressors where losses through imperfections and improper working conditions range from 15 to 25 per cent, while under favorable conditions and with the average compressor, the loss averages from 8 to 12 per cent. So that to get sufficiently accurate results in finding capacity of the compressor, subtract 12 per cent from above computation, which gives nearly accurate figures. The following table will be found useful for quickly ascertaining the capacity of an air compressor, also to find the cubical contents of any cylinder, receiver, etc. The first column

is the diam. of cylinder in inches. The second shows the contents in cubic feet, for each foot in length.

Contents of a Cylinder in Cubic Feet for Each Foot in Length.

Diam. Inches.	Cubic Contents.	Diam. Inches.	Cubic Contents.	Diam. Inches.	Cubic Contents.	Diam. Inches.	Cubic Contents.	Diam. Inches.	Cubic Contents.
1	.0055	6	.1963	11	.6600	20	2.182	36	7.069
1¼	.0085	6¼	.2130	11¼	.6903	20½	2.292	37	7.468
1½	.0123	6½	.2305	11½	.7213	21	2.405	38	7.886
1¾	.0168	6¾	.2485	11¾	.7530	21½	2.521	39	8.296
2	.0218	7	.2673	12	.7854	22	2.640	40	8.728
2¼	.0276	7¼	.2868	12½	.8523	22½	2.761	41	9.168
2½	.0341	7½	.3068	13	.9218	23	2.885	42	9.620
2¾	.0413	7¾	.3275	13½	.9940	23½	2.885	43	10.084
3	.0401	8	.3490	14	1.069	24	3.012	44	10.560
3¼	.0576	8¼	.3713	14½	1.147	25	3.142	45	11.044
3½	.0668	8½	.3940	15	1.227	26	3.400	46	11.540
3¾	.0767	8¾	.4175	15½	1.310	27	3.687	47	12.048
4	.0873	9	.4418	16	1.396	28	3.976	48	12.566
4¼	.0985	9¼	.4668	16½	1.485	29	4.587	..	......
4½	.1105	9½	.4923	17	1.576	30	4.909	..	......
4¾	.1231	9¾	.5185	17½	1.670	31	5.241	..	......
5	.1364	10	.5455	18	1.767	32	5.585	..	......
5¼	.1503	10¼	.5730	18½	1.867	33	5.940	..	......
5½	.1650	10½	.6013	19	1.969	34	6.305	..	......
5¾	.1803	10¾	.6303	19½	2.074	35	6.681	..	......

To find the capacity of an air-cylinder, multiply the figures in the second column by the piston travel in feet per minute. This applies to double-acting air cylinders. In the case of single-acting air cylinders, the result should be divided by 2.

THE McKIERMAN DRILL COMPANY'S AIR COMPRESSOR.

The air-cylinder and water-jacket are one complete casting. The heads are made with hoods and provision made for cool air in-take.

The atmosphere valves are bronze, of poppet form. Therefore, there is no vacuum and the cylinder fills absolutely with free air. The valves are closed by mechanical means.

The discharge valves are self-acting, are made of bronze. All of them are free to inspection without removal or disturbance of other parts.

The cooling apparatus, or heat-preventing device, is extremely effective. Water jacket completely surrounds the cylinder, water

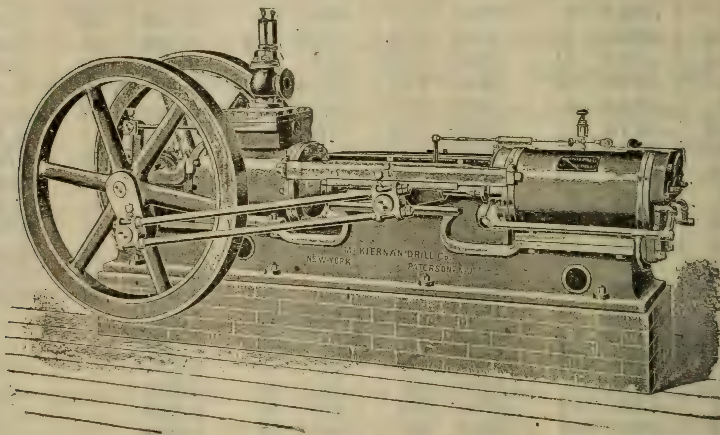


Fig. 344. Horizontal single stage compressor.

is forced to wash the walls and is kept in rapid motion from bottom to top, from end to end, absorbing heat rapidly. It enters the jacket at bottom, flows from end to end, around partitions, back and forth and up. Follows natural laws in absorbing, retaining and dispelling the heat of air.

Regulation of pressure and speed is entirely automatic. The regulating device is one of those in which the air weighs the steam admitted to the cylinder. Throttle may be thrown wide open at start, then the regulator takes absolute control, governing the speed from highest to lowest rate, varying the speed for

variable amounts of air which may be required and in such manner as to keep the pressure constant.

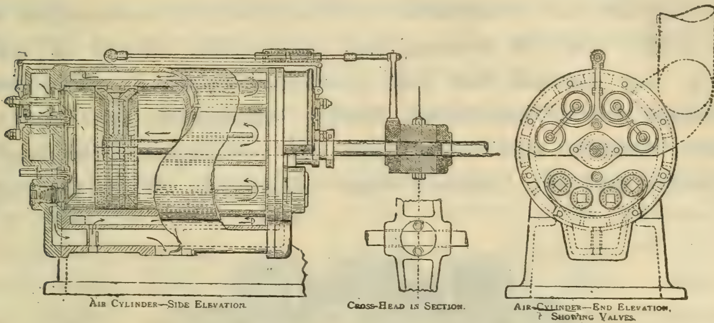


Fig. 345. The Bennett automatic air compressor.

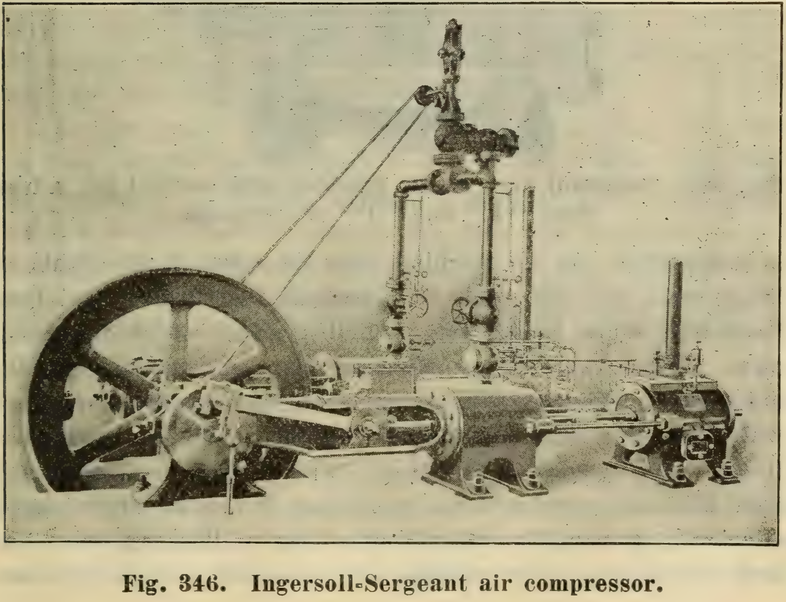


Fig. 346. Ingersoll-Sergeant air compressor.

INGERSOLL-SERGEANT AIR COMPRESSOR.

This engine, illustrated in Fig. 346, is fitted with Ingersoll-Sergeant air compressor cylinders, and in connection with the Pohlé air lift system, has double the supply of water, using only one-half the fuel previously required. The steam cylinders are of the duplex Corliss condensing type and connected tandem, and on each side are two Ingersoll-Sergeant air cylinders and two Conover water cylinders. When the engine

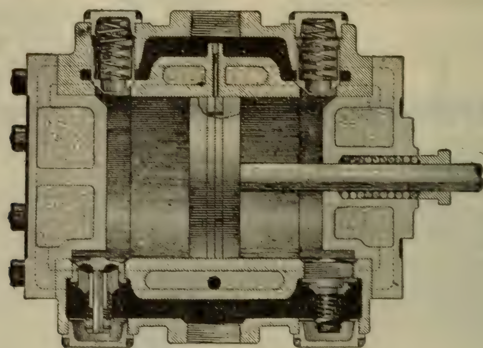


Fig. 347. Sectional view of air cylinder with vertical lift valves, class "E" and "F" compressors.

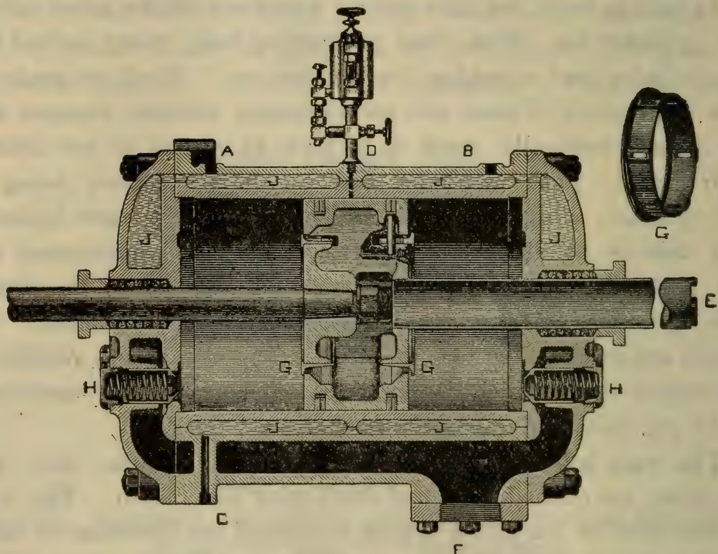
is in operation, the air cylinders raise the water by the Pohlé air lift system, from the wells to a tank at the surface, and from there it is taken by the water cylinders and forced to the standpipe. The cost of this combination compares favorably with the old plan of using separate compressors and water pumps, each with their own steam cylinders, and the saving in attendance, friction and foundation commends its use. The engines run at a fixed moderate speed and the regulation of the air and water is effected by passing the water from suction to discharge when the tank is too low, and by mechanically unloading the air cylinders

with a pressure regulator when the tank is too full. The regulation is done mechanically, with floats at the top and bottom of the receiving tank. This combination can also be furnished with straight line compressors; the advantage of the duplex is that should it be necessary, the one side of the engine can be disconnected and the other side made to do the work.

As will be seen, the inlet valves, which are on the lower side of the cylinder are offset, thus preventing their being sucked into the cylinder and wrecking the compressor. They are made out of a solid piece of steel and are extremely durable, because they are placed vertically, work in a bath of oil and do not slide on their seats. Both the inlet and discharge valves, being in the cylinder, allow the heads to be thoroughly water-jacketed, and this is an important feature when it is remembered that the heat of compression is greatest at the end of the stroke. The cylinder is, therefore, completely water-jacketed. The valves are arranged so that the air can be taken from outside of the engine room, which increases the efficiency of the machine 8 to 15 per cent, and are easily accessible.

The two inlet valves are located in the piston, and, with the tube, are carried back and forth with the piston. The valve on that face of the piston which is toward the direction of movement is closed, while the one on the other face is open. This is exactly as it should be in order to force out the compressed air from one end of the cylinder while taking in the free air at the other; when the piston has reached the end of its travel there is, of course, a complete stop while the engine is passing the center, and an immediate start in the other direction. The valve which was open immediately closes. There is no reason for its remaining open any longer, and it closes at exactly the right time, its own weight being all that is necessary to move it. The valve on the other side is left behind by the piston and the free air is admitted to that end of the cylinder for compression on the

return stroke. No springs are used, and there is none of the throttling of the incoming air, and none of the clattering or hammering so noticeable with poppet-valves. As there is nothing to make the valve move faster than the piston, it stays behind until the piston stops, leaving the port wide open for the admission



DETAILS OF PISTON INLET AIR CYLINDER.

A.—Circulating Water Inlet. D.—Oil Hole for Automatic Oil Cup. G.—Piston Inlet Valv
B.—Circulating Water Outlet. E.—Air Inlet (through piston inlet pipe). H.—Discharge Valve
C.—Water Jacket Drain Pipe. F.—Air Discharge (showing flange). J.—Water Jacket.

Fig. 348. Sectional view of Ingersoll-Sargeant single compressor.

of air. It is well known that while the fly-wheel and, of course, the crank, rotate at a uniform speed, the movement of the piston is not uniform, but gradually increases in speed from the start till the crank has reached half-stroke, when it gradually slows up till the crank is on the center, and at this moment of full stop the valve gently slides to its seat.

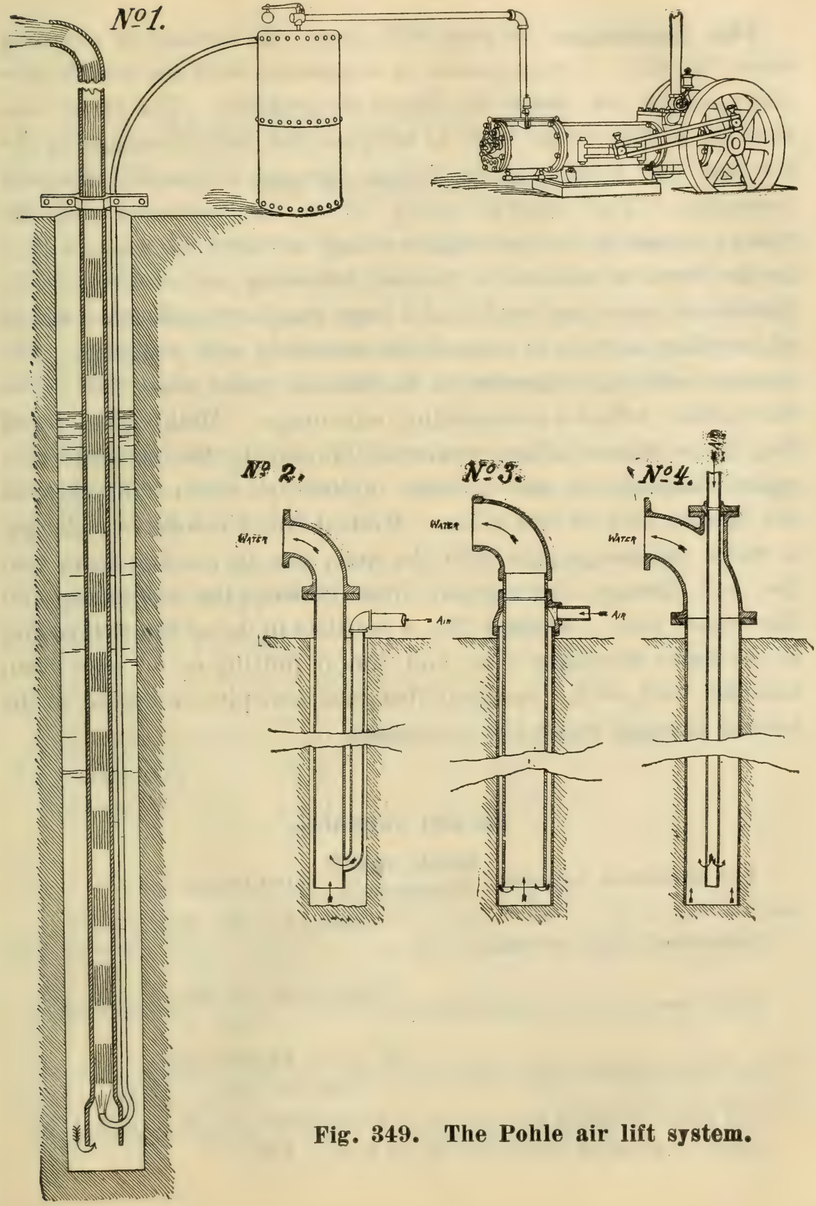


Fig. 349. The Pohle air lift system.

The illustrations on page 807 show the method of pumping water by air. A compressor in connection with the air-lift system pumps the water by direct air pressure. The pump consists of a water pipe and an air pipe, the latter discharging the air into the former at its bottom, through a specially designed foot-piece. The natural levity of the air compared with the water, causes it to rise and, in rising, to carry the water with it in the form of successive pistons, following one another. This system of pumping has found a large range of application and is of peculiar service in connection with deep well pumping. For this purpose, the absence of mechanical parts many feet below the surface, offers a commanding advantage. Method No. 1 and No. 2 are almost alike, consisting in placing the air and water pipes alongside of one another in the well, connecting them at the bottom with an end piece. Method No. 3 consists in placing a water discharge pipe into the well; the air passing down into the well through the annular space between the well casing and the water pipe. Method No. 4 consists in using the well casing as the water discharge pipe, and simply putting an air pipe down into the well, with a specially designed foot-piece attached at the bottom through which the air escapes.

Air Lift Formulas.

For maximum economy $\frac{\text{Height of Lift}}{\text{Submersion}}$ should equal 0.5

$$\text{Gallons of water raised per Min.} = \frac{125 \times \text{Cu. Ft. Free Air}}{\text{Lift in Feet}}$$

$$\text{Cubic feet of free air per Min.} = \frac{\text{Gals. raised per Min.} \times \text{Lift}}{125}$$

$$\text{Lift in feet above water level} = \frac{125 \times \text{Cu. Ft. Free Air}}{\text{Gals. per Min.}}$$

Air press. required to start lift = Submersion + Lift $\times$ .434 + 5

Ratio of areas of water pipe to air pipe = 6 to 1

CHAPTER XXVIII. — CONTINUED.

THE METRIC SYSTEM.

It frequently happens that an engineer, in reading books and papers devoted to steam engineering, is confronted with terms taken from the metric system, which he does not understand. We present a few of the metric system terms most commonly used, with their values in feet and inches, also, gallons, quarts, pounds, tons, etc.

A French meter is 39.37079 inches long, or a little less than $39\frac{3}{8}$ inches. It is generally taken, — for convenience in figuring, — at 39.37 inches.

1 decimeter is	$\frac{1}{10}$	of a meter, or,	3.937	inches	nearly.
1 centimeter is	$\frac{1}{100}$	" " "	.3937	"	"
1 millimeter is	$\frac{1}{1000}$	" " "	.03937	"	"

ALSO.

1 decameter equals	10	meters, or,	32.80	feet	nearly.
1 hectometer "	100	" "	328	"	"
1 kilometer "	1000	" "	3280	"	"

APPLICATION.

1. An engine shaft is 5 meters long, what is its length in feet and inches? Ans. 16 ft. $4\frac{3}{4}$ ins. nearly.

Operation: $\dfrac{39.37 \times 5}{12} = 16.4$ ft. nearly.

2. An engine cylinder is 10.3 decimeters in diameter, how much is this in inches? Ans. $40\frac{1}{2}$ ins. nearly.

Operation: $3.937 \times 10.3 = 40.55$ ins. nearly.

3. A piston-rod is 8.7 centimeters in diameter, how much is this in inches? Ans. $3\frac{3}{8}$ ins. nearly.

Operation: $.3937 \times 8.7 = 3.42$ ins. nearly.

4. A chimney is 5.1 decameters tall, how much is this in feet and inches? Ans. 167 ft. 3 ins. nearly.

Operation: $32.80 \times 5.1 = 167.28$ ft.

5. How many miles are there in 30.2 kilometers?

Ans. $18\frac{7}{10}$ miles nearly.

Operation: There are 5280 ft. in a mile.

$$\text{Then, } \frac{3280 \times 30.2}{5280} = 18.7 \text{ miles.}$$

6. A valve has 2 millimeters lead, how much is this in fractional parts of an inch? Ans. $\frac{5}{64}$ in. nearly.

Operation: $.03937 \times 2 = .07874$.

And, $.07874 \times 64 = \frac{5}{64}$ nearly.

7. How many square feet in a circle whose diameter is one meter? Ans. $8\frac{1}{2}$ nearly.

$$\textbf{Operation:} \frac{39.37 \times 39.37 \times .7854}{144} = 8.45.$$

8. The cylinder clearance is 1.1 cubic decimeter, how many cubic inches in the clearance? Ans. 67 nearly.

Operation: $3.937 \times 3.937 \times 3.937 \times 1.1 = 67.12+$

ALSO.

1 gramme equals 15.433 grains, or 1 ounce nearly.

1 kilogramme equals 2.2047 pounds avoirdupois.

1 tonne equals 1.1024 tons of 2000 lbs.

ALSO.

1 litre equals 1.0566 quarts.

CONSEQUENTLY,

1 U. S. gallon equals 3.79 litres nearly.

1 U. S. pint equals .4732 litres nearly.

1. A main shaft weighs 800 kilogrammes, how much is this in avoirdupois pounds? Ans. $1763\frac{3}{4}$ lbs. nearly.

Operation: $2.2047 \times 800 = 1763.76$.

2. An engine weighs 12 tonnes, how much is this in U. S. tons of 2000 lbs. each? Ans. $13\frac{1}{4}$ tons nearly.

Operation: $1.1024 \times 12 = 13.2288$.

3. A tank contains 9000 litres of water, how much is this in U. S. gallons? Ans. 2377.35 galls.

Operation: $\frac{1.0566 \times 9000}{4}$ Because 4 quarts equal 1 gallon.

THERMOMETERS.

In the U. S. the Fahrenheit scale is the one in most common use, although in our laboratories and for scientific purposes it is displaced by the Reaumer and Centigrade scales. Fahrenheit's scale marks the boiling point by 212 degrees, and the freezing point by 32 degrees above zero.

The Reaumer scale marks the boiling point by 80 degrees, and the freezing point by zero.

The Centigrade, or Celsius scale, marks the boiling point by 100 degrees, and the freezing point by zero. So that, reckoning from the freezing point of Fahrenheit, 180 degrees Fah. equal 80 degrees Reaumer, and 100 degrees Centigrade. Bearing in mind that Fahrenheit's zero is 32 degrees below the freezing point, one scale may readily be converted into another.

To convert degs. of Reaumer into those of Fah.

Rule. — Multiply by 9, divide by 4, and add 32.

Example: 80 degs. Reaumer equals how many degs. Fah? Ans. 212.

Operation: $80 \times 9 = 720$.

$$\text{And, } \frac{720}{4} = 180. \quad \text{Then, } 180 + 32 = 212.$$

To convert the degs. of Centigrade into those of Fahrenheit.

Rule. — Multiply by 9, divide by 5, and add 32.

Example: 100 degs. Centigrade equal how many degs. Fah.? Ans. 212.

Operation: $100 \times 9 = 900$.

$$\text{And, } \frac{900}{5} = 180.$$

Then, $180 + 32 = 212$.

So, also, 3 degs. Centigrade equal 37.2 degs. Fahrenheit.

Thus: $3 \times 9 = 27$. And, $\frac{27}{5} = 5.2$. Then, $5.2 + 32 = 37.2$.

ROPE TRANSMISSION.

There are two systems of rope transmission, the English, or multiple-rope system, and the American or continuous wound rope system in which the necessary adhesion of rope to sheave is obtained by a tension carriage. We will treat of the American system only, as it is almost universally used in this country to the exclusion of the other. One of the most common mistakes is to lead the rope to the tension carriage from the tight or pulling side of the drive, and putting on an abnormal amount of tension weight in a vain endeavor to take out the slack. Under the enormous strain of such an arrangement the rope wears out very rapidly, and more frequently parts at the splice. It is desirable in all cases of rope transmission to so arrange the drive that the slack side of the rope shall be on the upper part of the pulley

thus increasing the arc of contact, as the two sides will then approach each other when in motion. The working strain in pounds on a rope should not exceed 200 times the square of the diameter of the rope. The speed of the rope should not exceed 5500 feet per minute, and this speed gives the best results in H. P. The practical limit to the number of ropes for one sheave cannot be definitely named. The only limiting condition is the ability of the tension carriage to keep up the slack and when the number of ropes exceeds the capacity of one carriage, a second may be added and the drive made double. Diameters of sheaves should not be less than 40 diameters of the rope, and 50 to 60 diameters are advisable, being justified by greater length of life of the rope.

HORSE POWER TRANSMITTED BY ROPES.

The following table gives the horse-power transmitted by a single manila rope when the arc of contact is not less than 165 degrees, and the tension not greater than 200 times the square of the diameter of the rope.

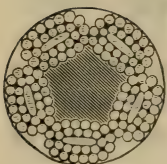
Velocity of Rope in Feet per Minute.	Diameter of Rope.						
	5/8″	3/4″	1″	1¼″	1½″	1¾″	2″
1000	1.24	2.25	3.57	5.59	8.02	10.85	14.20
2000	2.70	3.84	6.84	10.68	15.39	20.93	27.36
2500	3.30	4.71	8.38	13.10	18.86	25.66	33.54
3000	3.83	5.46	9.80	15.39	21.87	29.74	38.88
3500	4.30	6.23	11.09	17.33	24.94	34.03	44.35
4000	4.74	6.83	12.15	18.98	27.33	37.17	48.59
4500	5.01	7.24	12.89	20.15	29.00	39.45	51.57
5000	5.20	7.47	13.29	20.76	29.89	40.65	53.15
5500	5.29	7.60	13.53	21.14	30.43	41.39	54.11
6000	5.08	7.32	13.10	20.36	29.32	39.77	52.12
6500	4.74	6.83	12.13	19.00	27.34	37.21	48.63
7000	4.12	5.93	10.54	16.47	23.72	32.26	42.18
7500	3.25	4.67	8.32	13.00	18.73	25.42	33.23

TO TEST THE PURITY OF ROPE.

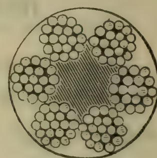
A simple test for the purity of manila or sisal rope is as follows: —

Take some of the loose fiber and roll it into balls and burn them completely to ashes, and, if the rope is pure manila, the ash will be a dull grayish black. If the rope be made from sisal the ash will be a whitish gray, and if the rope is made from a combination of manila and sisal the ash will be of a mixed color.

WIRE ROPE DATA.



HOISTING ROPE.



PATENT FLATTENED STRAND.

DIA. IN INCHES.	HERCULES. Price per foot in cents.	HERCULES. Breaking strain in tons of 2,000 lbs.	CRUCIBLE. Price per foot in cents.	CRUCIBLE. Breaking strain in tons of 2,000 lbs.	IRON. Price per foot in cents.	IRON. Breaking strain in tons of 2,000 lbs.
½	19½	13.5	14½	9	10½	4
⅝	26½	22 5	18¼	15	15½	6
¾	35	32	24	21	21	9
⅞	45	40.5	30	29	26	13
1	56½	56	39½	38	34	17
1⅛	68	67	50	47	43	21
1¼	82	84	59⅛	56	52	28
1½	123	124	86	81	74	40
1¾	173	168	121	109	104	54
2	202	211	144	140	120	66
2¼	257	260	182	176	152	75

19 WIRE ROUND STRAND.

DIA. IN INCHES.	HERCULES. Price per foot in cents.	HERCULES. Breaking strain in tons of 2,000 lbs.	CRUCIBLE. Price per foot in cents.	CRUCIBLE. Breaking strain in tons of 2,000 lbs.	IRON. Price per foot in cents.	IRON. Breaking strain in tons of 2,000 lbs.
½	16½	12.5	11	8.8	8	4
⅝	22½	20	14	13 6	12	6
¾	30	29	18	19.4	16	9
⅞	39	36	23	26	20	13
1	48½	50	30	34	26	17
1⅛	57½	60	38	42	33	21
1¼	71	77	46	50	40	25
1½	103	113	66	72	57	36
1¾	147	157	93	96	80	48
2	172	191	111	124	92	62
2¼	218	238	142	156	117	74

ALTERNATING CURRENT MACHINERY.

CHAPTER XXIX.

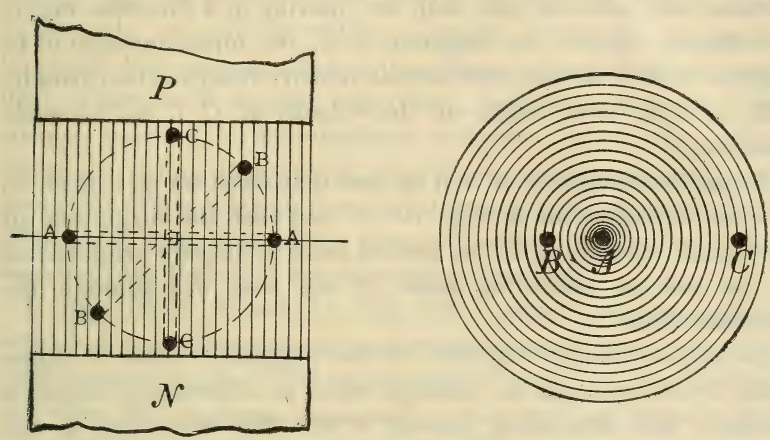
THE PRINCIPLES OF ALTERNATING CURRENTS.

The actions of alternating currents are not so easily understood as those of continuous currents and to most men not familiar with the subject they appear to be a mystery that can only be fathomed by those who are well versed in the higher branches of mathematics. As a matter of fact, when we once get on the right track, alternating current actions present no more difficulty to the man of fair mental ability, who is willing to work to learn, than the more simple continuous current actions. What makes alternating currents difficult to understand is, that in consequence of the ever-changing strength of the current, inductive actions are developed that react upon the current itself so that it becomes impossible to determine the magnitude of the current, the e.m.f. or the energy flowing in the circuit by the simple rules used for continuous currents. As the strength of an alternating current is constantly changing the magnitude of the inductive actions is constantly changing, and this fact further increases the difficulty of the subject.

In studying the principles of continuous currents we learn that when a conductor is moved across a magnetic field an e.m.f. is developed in it; and thus we understand the operation of a generator, as we know that when the armature revolves, it carries the conductors upon its surface through the magnetic flux that issued from the poles of the field. We further learn that inas-

much as the magnitude of the e.m.f. is increased by increasing the strength of the magnetic flux, or the number of conductors on the armature or the velocity of rotation, that one or all these factors must be increased to increase the voltage. Thus we come to consider that to induce a high e.m.f. we must have a strong magnetic field. Now one of the first things that the student of alternating currents finds out is that in an alternating current circuit, the strongest e.m.f. induced by the action of the current itself, comes at the very time when the magnetic field is the weakest, and this appears to him to completely upset all the principles of continuous currents; but in reality it does not. To be able to get over this stumbling block successfully it is necessary to realize that the magnitude of the e.m.f. induced in a conductor that is moved through a magnetic field is not dependent upon the strength of the magnetic field, but upon the rate, or rapidity with which the conductor cuts the magnetic flux. Now it so happens that in a continuous current generator, the rapidity with which the conductors cut the magnetic flux increases with increase in the strength of the magnetic field, or the velocity of rotation, and thus it comes about that in this case, the increase in the induced e.m.f. appears to be due to increase in armature velocity or field strength when in reality it is due to increase in the rate at which the conductors cut through the magnetic flux. The magnetic flux developed by an alternating current alternates precisely as the current does, and, as will be clearly explained presently, this magnetic flux cuts through any conductors in its path, and the rate at which it cuts them is the greatest at the instant when the direction of the flux is changing, and this is the instant when the flux is nothing, so that the e.m.f. induced by the magnetic flux developed by an alternating current is the greatest at the very instant when the magnetic field has a zero strength. The foregoing facts can be made more clear by reference to diagrams.

Fig. 350 is a simple diagram that can be taken to represent a generator, either of continuous or alternating currents. The dark circles *A A*, *B B* and *C C* represent the sides of three loops of wire which may be regarded as wound upon the surface of an armature.



Figs. 350 and 351. Principle of electric generator.

The vertical lines represent a magnetic flux passing between the field poles *P* and *N*. If the armature upon which the three loops are mounted is rotated, e.m.fs., will be induced in each one of the loops, but the magnitude of these e.m.fs. will not be the same. If we take the instant when the loops are in the position shown, the e.m.f. in *A A* will be zero, while that in *C C* will be the highest and that in *B B* will be seven-tenths of that in *C C*. Now all these coils rotate at the same velocity being mounted upon the same armature, and all move through a magnetic field of the same strength, yet, in *A A* no e.m.f. is developed while in *B B* the e.m.f. is only seven-tenths of that developed in *C C*. The question is, why this difference? The answer is, that while loops *A A* move just as fast as *C C* they do not cut the magnetic flux

because they are moving in a direction parallel with the lines of force, the vertical lines, hence, the rate at which the magnetic flux is cut by them is zero, therefore the e.m.f. developed is zero. In *B B* the e.m.f. is seven-tenths of that developed in *C C*, because the sides of this loop are moving in a direction that is not directly across the magnetic flux, but forms an angle of 45 degrees with it, so that their actual velocity in a direction parallel with *A A* is seven-tenths of the velocity of *C C* in this same direction.

From the foregoing it will be seen that when we get down to a close examination of Fig. 350 we find that the magnitude of the e.m.f. developed in the several loops is directly proportional to the rate at which the sides of the loop cut through the magnetic flux.

Let us now consider Fig. 351. In this diagram, circle *A* represents a wire, seen end on, through which an alternating current is flowing. An alternating current is one that flows first in one direction, and then in the opposite direction, and continues changing the direction in which it flows at regular intervals of time. Now it is self-evident that if a current flows through a wire in alternate directions, it must stop flowing in one direction before it can flow in the opposite direction, that is at the instant when the direction of flow is changing, there can be no current. Such being the case, when the current begins to flow in either direction, it must increase in strength gradually up to a certain point, and then begin to decrease, so as to reduce to nothing at the instant when the direction of flow changes. As is explained in the section on continuous currents, when a current of electricity flows through a wire, a magnetic flux is developed around the wire and this can be represented by lines of force drawn in the form of circles, as in Fig. 351. If there is no current flowing through the wire there is no magnetic flux, therefore, if we consider the instant when a current begins to flow, we can imagine

that at this instant the magnetic flux begins to expand outward from the wire, and since the circular lines are drawn to represent this flux we can assume that these expand outward, like the ripples on the surface of a pond when a pebble is thrown into the water. So long as the current flowing through the wire increases in strength, just so long will the magnetic circles of force expand, but when the current reaches its greatest strength the circular lines of force will become stationary, and will remain so if the current remains at its maximum strength; but if the current begins to reduce in strength as soon as it reaches its maximum, then the circular lines of force will begin to contract immediately after they stop expanding, just as a swing will begin to move backward the instant it stops swinging forward.

If the circles *B* and *C* in Fig. 351 represent two wires parallel with *A*, it is evident that the magnetic circles of force when they move outward from *A* will cut through *B* and *C* in one direction, and when they contract back upon *A* they will cut through these two wires in the opposite direction. When these circular lines of force cut through the wires *B C* they will induce e.m.fs. in the latter, and if these e.m.fs. are positive when the lines of force expand, they will be negative when the lines contract. When the current reaches its maximum strength and the circular lines of force become stationary for an instant, they will not cut the wires *B* and *C* and at this instant there will be no e.m.f. induced in these wires. Now the circular lines of force become stationary at the very instant when the current flowing through the wire reaches its greatest strength and is on the point of reducing, so that at this instant the e.m.f. induced in the wires *B* and *C* is zero.

The highest e.m.f. induced in *B* and *C* occurs at the instant when the current flowing through *A* is changing its direction, or, in other words, at the instant when there is no current. Just before the current reduces to zero, the circular lines of force are contracting upon wire *A*, and the instant after the cur-

rent reduces to zero and changes its direction, these lines of force will be expanding so that in the first case the lines of force will sweep over wires B and C in a direction toward A, and in the second case they will sweep over these wires in a direction away from A. From this fact it might be inferred that the e.m.f. induced in the two cases would be in opposite directions, but this is not so, owing to the fact that the lines of force change in direction when the current changes, so that if while contracting they are directed clockwise, as soon as they begin to expand they will be directed counter clockwise. As a result of this change in the direction of the lines of force when they change from contracting to expanding, the e.m.fs. induced in B and C are in the same direction before the lines stop contracting and after they begin to expand. The circular lines of force stop contracting and begin to expand at the same instant, so that the inductive action developed by the contracting lines is followed up without a break by the expanding lines. In alternating currents such as are actually used in practice, the rate at which the strength of the current changes is the greatest when it is just beginning to grow, and when it is reduced almost to zero, and on this account the highest e.m.f. induced in wires B and C occurs at the instant when the direction of the current is changing, that is, when the current is zero. Alternating currents can be developed in which the rate of change in the current is not the greatest just when they begin to grow and when they are reduced nearly to zero and with such currents the highest e.m.f. induced in wires B and C would not come at the instant when the current is zero, but would come at the instants when the change in the current is the most rapid.

In every kind of alternating current, however, the instant when the e.m.f. induced in B and C is zero is the instant when the current reaches the maximum value, and begins to decrease, for this is the only instant when the circular lines of force are

immovable; it being the instant when they are about to change from expanding to contracting, while still flowing in the same direction. When the current becomes zero, the lines of force change from contracting to expanding but at this instant they also change their direction so that the new expanding circular lines of force take up the work if inducing an e.m.f. in the wires *B* and *C* at the very point where the contracting lines leave off.

The circular lines of force developed by the current flowing in *A*, cut through this wire as well as through *B* and *C*, hence, they induce an e.m.f. in *A*; that is an alternating current induces an e.m.f. in its own circuit as well as in adjoining circuits. The action upon adjoining wires is called mutual induction, and that upon its own wire is called self-induction. These e.m.fs. act in a direction opposite to that of the current that induces them.

The relations between alternating currents and e.m.fs. can be shown by means of diagrams, the simplest of which are con-

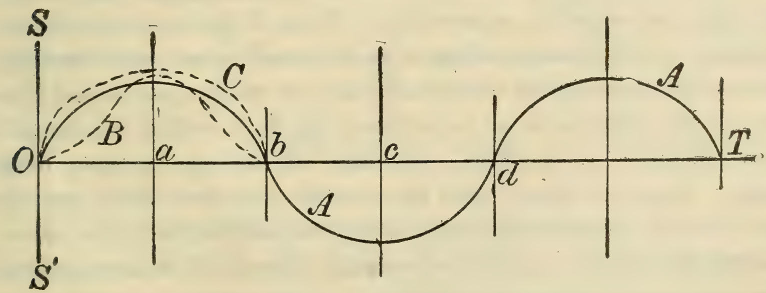


Fig. 352. Relation between current and electro-motive force.

structed in the manner shown in Fig. 352. In diagrams of this type the line *O T* represents time, thus if a point is assumed to move from *O* in the direction of *T* at a uniform velocity of say one foot per second, then a length of one inch will represent an interval of time of one-twelfth of a second. Distances measured in the vertical direction, along *O S* represent the magnitude of

the current or e.m.f. Positive currents and e.m.f. are indicated above the time line $O\ T$ and negative currents and e.m.fs. below this line. Thus the wave line $A\ A\ A$ can represent an alternating current or e.m.f. or an alternating magnetic flux. This curve it will be seen is above $O\ T$ from O to b, and below $O\ T$ from b to d, being again above from d to T. The two sections of the curve from O to d constitute one cycle, or two alternations. The portions between the lines $O\ a$, $a\ b$, $b\ c$, $c\ d$ are called quarter cycles or quarter periods. The time from O to d is called one period, and if this is equal to one-tenth of the distance that represents one second, then there are ten periods to one second. This fact is indicated by saying that the periodicity of the current is ten, or that its frequency is ten. The frequency of alternating currents in common use ranges between 20 and 130.

The curve $A\ A$ in Fig. 352 represents a current or e.m.f. that increases or decreases at a certain rate, but for a current varying at some other rate it would be necessary to use a curve of different shape to correctly represent it. Thus if the current does not increase so fast when rising from the zero value, but increases faster when nearing its maximum value we will require a modification of the curve such as is indicated by B, in which the slope is more gradual on the start, and near the middle becomes more steep. If on the other hand the current increases more rapidly on the start, and less rapidly as it approaches the maximum value, we will have to use a curve something like C which is steeper at the ends and flatter at the middle.

The actual form of curve required to correctly represent an alternating current depends upon the rate at which the current varies, and this rate depends upon the construction of the machine in which it is generated. For the purpose of simplifying calculations it is necessary to assume that the rate of variation of a current is such that it can be represented in a diagram such as Fig. 352 by some form of curve that can be drawn in accordance with

some fixed rule. The curve *A A* is of circular form, but there are few alternating current generators that develop currents that such a curve can properly represent.

If a current alternates in equal intervals of time, and the rate of variation is the same when it is flowing negatively as when it is flowing positively, then it can be represented by a curve that is of symmetrical construction, such as *A A* in which the intervals of time *O b*, *b d* are equal and the curves above the line *O T* are of the same shape as those below it. Such a current is called a symmetrical periodic current, and it is the only kind with which we have to do in practice. It can be readily understood, however, that the current can be far from regular, that is, the time during which it flows positively can be more or less than the time during which it flows negatively, and the rate of variation in the two

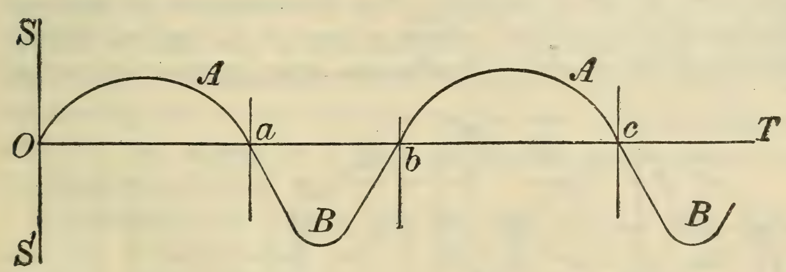


Fig. 353. Irregular periodic curve.

instances can be different. The curves in Figs 353 and 354 illustrate currents of this kind. In Fig. 353 the positive impulses of the current are longer than the negative, as is shown by the greater length of lines *O a*, *b c* as compared with *a b*. It will also be seen that the rate of variation is different as is indicated by the difference in the form of the portions *A A* and *B B* of the curve. In Fig. 354 the irregularity is still greater, as all the time intervals *O a*, *a b*, *b c*, *c d*, are different, as are also the portions *A B C D E* of the curve.

The alternating currents developed by alternating current generators have such a rate of variation that they can be represented in diagrams by means of what is known as a sine curve.

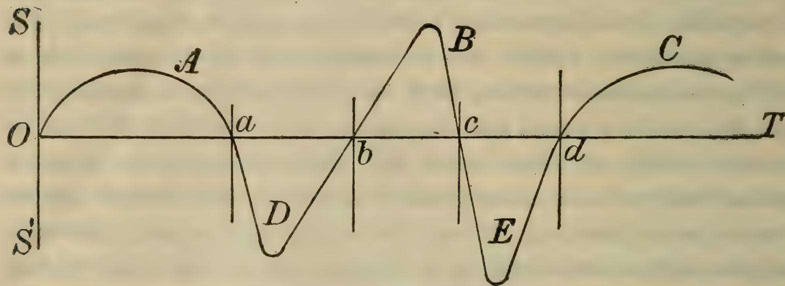


Fig. 354. Showing still greater irregularity.

This curve is not a perfectly true representation of practical alternating currents, but it comes so near to it that calculations based upon the assumption that the sine curve represents the actual

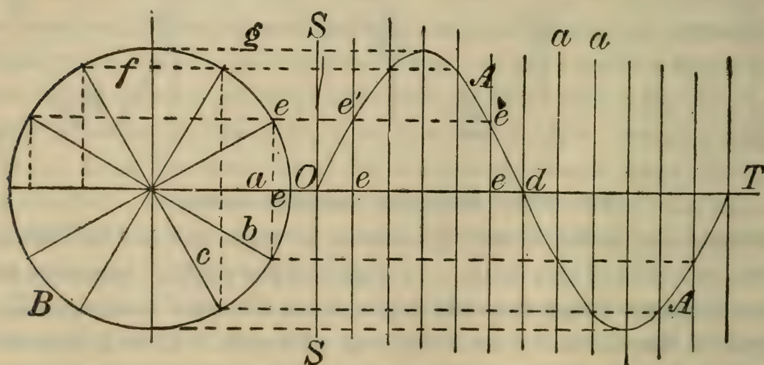


Fig. 355. Construction of sine curve.

variation, do not depart from the truth by more than two or three per cent, and in some cases less than that. As the sine curve is commonly used to represent alternating currents we will show

how it is constructed by the aid of Fig. 355. In this diagram diametrical lines *a b c* are drawn on the circle *B*, dividing it into any desired number of equal parts. A distance *O T* on the horizontal line is divided into an equal number of equal parts and perpendicular lines *a a* are drawn at these divisions. From the points where the lines *a b c* cut the circle lines are drawn parallel with *O T* as shown at *e f g* and the points where these cut the corresponding perpendicular lines *a a* form points of the sine curve *A A*. The distance *O T* can be made anything desired without affecting the character of the curve, the only difference being that if it is short the curve will be more pointed than if it is long.

One reason why it is assumed that alternating currents vary in accordance with a sine curve is that if the variation is at this rate the e.m.f. induced by the magnetic flux developed by the current will also vary in accordance with the sine curve, so that the current, the magnetization and the induced e.m.f. can be represented by sine curves, and thus the process of calculating the effect of the induced e.m.f. upon the strength of the current can be greatly simplified.

By looking at Fig. 350 it can be seen at once that if the loop *A A* is revolved at a uniform velocity, and the magnetic field between the poles *P* and *N* is of uniform strength at every point, the e.m.f. induced in *A A* will vary in strict accordance with the variations of the sine curve *A A* of Fig. 355, for in the position *A A* the e.m.f. will be zero, and in position *C C* it will be the maximum, while in any intermediate position such as *B B* it will be equal to the actual velocity of the sides of the loop measured in the direction parallel with *AA* , and this velocity is equal to the distance of the side of the loop from the horizontal line *A A*. Now the height of the sine curve *A A* in Fig. 355 at any point is also equal to the distance from the end of the corresponding line in circle *B* from the horizontal

line, that is, the distrance $e\ e'$ from the horizontal line to the curve is the same as the distance $e\ e$ on the circle.

The complete sine curve from O to T is traced by following the rotation of the radius of the circle through one complete revolution. On that account this distance $O\ T$ is taken to represent one revolution, and is divided into 360 degrees, the same as the circle. Half the distance, or $O\ d$, is equal to 180 degrees, and one-quarter the distance is 90. The vertical lines $a\ a$ in Fig. 355 are 30 degrees apart.

The way in which sine curves are used to represent alternating currents and e.m.fs. is shown in Fig. 356. In this diagram,

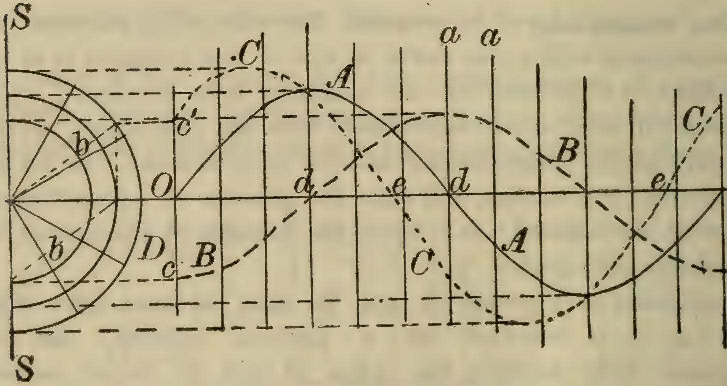


Fig. 356. Manner of indicating alternating currents.

let the curve A represent an alternating current flowing through a wire. As is fully explained in the foregoing, this current will develop an alternating magnetic flux, and this flux will increase and decrease as the current increases and decreases, that is, it will keep in time with the current, or in step with it, as it is commonly expressed. Such being the case, the curve A can be used to represent the magnetic flux as well as the current, providing we assume a proper scale for both. Looking at the half circle to the left of the figure, it will be seen that curve A is described by

a radius rotating around the middle circle. Remembering what was said in connection with Fig. 351 as to the time relation between the magnetic flux and the e.m.f. induced thereby, we will realize that at the instant O when the flux is zero, the induced e.m.f. must be at the maximum value, and it will act in opposition to the e.m.f. that drives the current through the wire, hence, in the diagram, it will have to be drawn below line $O\ T$. Let the maximum value of this induced e.m.f. be equal to $O\ c$, then for all other values it will be correctly represented by the sine curve B, which is traced by the rotation of the radius of the inner circle.

At the instant of time O, the magnetic flux is zero, hence the radius of the middle circle from which curve A is traced must be in the direction of line $O\ T$. At this same instant the induced e.m.f. is at the maximum value hence the radius that traces curve B must be in the vertical position parallel with $O\ c$. From this we see that in relation to time the curves A and B that represent the magnetic flux and the induced e.m.f. are one-quarter of a cycle apart, that is the induced e.m.f. is 90 degrees behind the magnetization, and also 90 degrees behind the current that flows through the wire.

No kind of electric current, whether continuous or alternating, can flow through a circuit unless there is an e.m.f. to drive it, and this e.m.f. must be sufficient to impel the current against all resistances of any kind that it may encounter. The e.m.f. that impels a current through an alternating current circuit is called the impressed e.m.f. In Fig. 356 it is evident that the impressed e.m.f. must be sufficient not only to overcome the actual resistance that opposes the flow of the current represented by curve A, but also sufficient to overcome the opposing action of the induced e.m.f. represented by curve B. Now the e.m.f. required to overcome the resistance that opposed the flow of the current can be represented by the curve A, in precisely the same

way as this curve represents the magnetization; hence, the curve C which represents the impressed e.m.f. must at every point be equal, in height, from the line $O\ T$, to the sum of the heights of the curves A and B, when these two curves are on opposite sides of $O\ T$, or to their difference when they are on the same side. At the instant O it is clear that as the current is zero, the impressed e.m.f. C must be of the value $O\ c'$ to balance the induced e.m.f. B for if it were not, there would be a current flowing negatively under the influence of e.m.f. B. At any instant between C and d, the impressed e.m.f. C must be equal to the sum A and B, that is, the distance from C to the time line $O\ T$ must be equal to the distance between the curves $A\ B$ measured on the same vertical line. At the instant d the induced e.m.f. is zero, hence the impressed e.m.f. is equal to the distance of curve A above line $O\ T$. For any interval of time between d and e, the impressed and the induced e.m.fs. are acting together, so that the first named, that is, curve C, need only be equal to the difference between A and B.

By studying the diagram Fig. 356 it will be seen that the curve C, which represents the impressed e.m.f., is described by the rotation of the radius of the outer circle at D, and in order that this e.m.f. may have the value of $O\ c'$ at the instant O, it is necessary for the describing radius at this instant to be in the position b. From this it will be seen that the impressed e.m.f. is not in time with the current but in advance of it by a time interval that is equal to the angle formed by the radius b with the line $O\ T$.

If two alternating currents, e.m.f. or magnetic fluxes are in time with each other they are said to be in phase, but if they are not in time they are out of phase. In Fig. 356 the current, the impressed e.m.f. and the induced e.m.f. are out of phase with each other. The impressed e.m.f. leads the current, and the latter leads the induced e.m.f. This relation is also expressed by say-

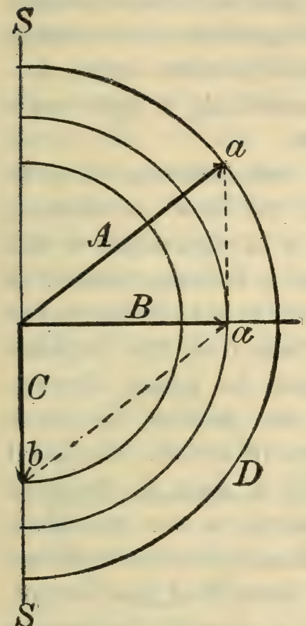
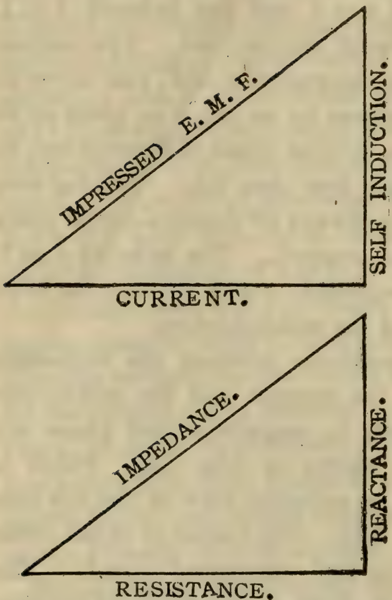


Fig. 357. Enlarged View of Fig. 356.

ing that the current lags behind the impressed e.m.f. and the induced e.m.f. lags behind the current. The current and the impressed e.m.f. can never be out of phase by an angle as great as 90 degrees, but the phase difference can be any angle less than this. The induced e.m.f. is always 90 degrees out of phase with the current. The induced e.m.f. in the circuit in which the current flows is called the self-induction.

The relations between the impressed e.m.f., the current and the self-induction both in magnitude and phase are clearly shown in Fig. 357, which is simply an enlarged view of the left side of Fig. 356. The radius *A* of the outer circle is the impressed e.m.f. The radius *B* of the middle circle is the current, and the radius *C* of the inner circle is the self-induction. The magnitude of any one of these three quantities at any instant of time is equal to the distance from the end of the line to the horizontal line. The radius *B* which represents the current is on the horizontal line, hence the current at the instant represented by the diagram is zero. The self-induction *C* has a value at this instant equal to the length of the line, that is, it is at the maximum value, and as it is below the horizontal line it is negative. The impressed e.m.f. *A*, has the value of *a a*, and being above the horizontal line, it is positive.

The phase relation and also the magnitude of these quantities is also shown in Fig. 358, which is constructed from Fig. 357 by removing the self-induction to the position of line *a a*. From Fig. 358 it



Figs. 358 and 359. Illustrating resistance, reactance and impedance.

can be seen that if we know two of the quantities we can always determine the other one by simply constructing a right angle triangle.

The self-induction acts to oppose the flow of current, hence it is equivalent to the addition to a certain amount of resistance to the circuit, but as can be seen from the diagrams it cannot be added directly, after the fashion in which numbers are added. To add it properly it must be placed at right angles to the resistance.

If the self-induction is divided by the strength of the current, we get a quantity that can be compared with the resistance, and this quantity is called the reactance and is measured in ohms precisely as the resistance is.

The flow of current in a continuous current circuit is opposed by the resistance only, but in an alternating current circuit it is opposed by the resistance and the reactance and the combined effect of these two is called the impedance of the circuit.

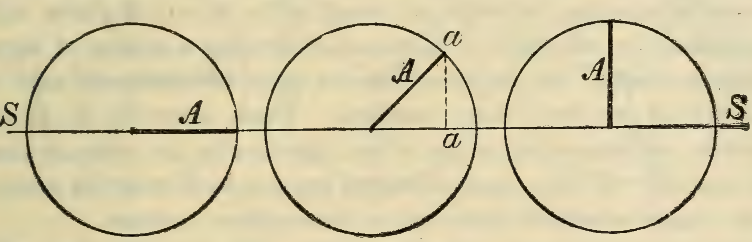
The relation between resistance, reactance and impedance is the same as that between impressed e.m.f., current and self-induction, and is shown in Fig. 359.

The reactance multiplied by the current gives the self-induction.

The impedance multiplied by the current gives the impressed e.m.f.

The resistance multiplied by the current gives the e.m.f. in phase with the current, which is also called the active e.m.f.

A sine curve diagram, such as is shown in Fig. 356, serves very well to enable the learner to understand the relation between the current and e.m.fs. but when this relation has been fully mastered, what is known as a clock dial diagram becomes more convenient, especially if we desire to represent several currents and their e.m.fs. Fig. 357 is virtually one-half of a clock dial diagram. A regular clock dial diagram to represent a single alternating current is shown in Figs. 360 to 362. The radius A represents the current, and is

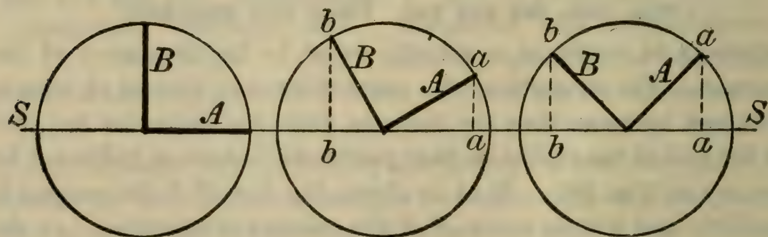


Figs. 360, 361 and 362. Clock dial diagrams.

supposed to rotate at a velocity equal to the frequency of the current. The strength of the current for any instant of time is obtained by measuring the distance from the horizontal line S S to the end of the radius at that particular instant as indicated by line a a in Fig. 361. If A is above the line S S the current is positive, and if it is below S S the current is negative. At the instant when A is in the vertical position, as in Fig. 362, the current is at its maximum value, and when A is horizontal as in Fig. 360 the current is zero. If we desire to find the relation between the current and impressed e.m.f. or the self-induction, we draw radial lines of the proper length to represent these e.m.fs. and in the proper angular position with reference to the current and then assume them to be locked together when they

are rotating so that the distances from the ends of each one to the line S S at any instant gives the values of the quantities at this instant.

Diagrams of this type are specially valuable for the representation of polyphase currents. Currents of this type are commonly spoken of as a two-phase current, or a three-phase current, or a polyphase current. Now there are no multiplephase currents. What is improperly called a two-phase current is a combination or two simple alternating currents so timed that they are out of phase with each other by one quarter of a period, or revolution. This constitutes a system of two-phase currents. Three simple alternating currents so timed as to be out of phase with each other by one-third of a period, constitute a system of three phase currents. In the first case we have two currents, and in the second we have three currents. These currents in either system are connected so as to act together in the same system of circuits. If the phase relations are not such as given above, they cannot constitute true, two or three-phase systems.

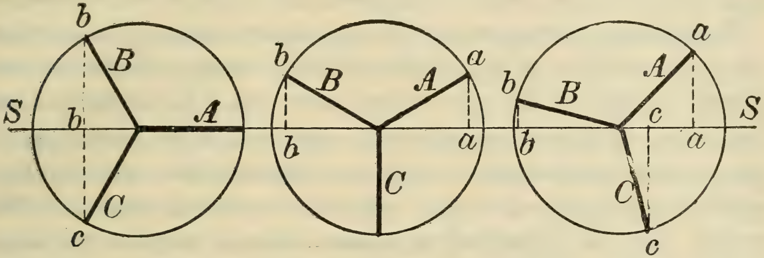


Figs. 363, 364 and 365. Phase relations for two-phase system.

The phase relations for the two-phase system are shown in Fig. 363 and for the three-phase system in Fig. 366. The two currents A B in Fig. 363 are at right angles with each other, and the three currents in Fig. 366 are 120 degrees apart, or one-third of a period, or cycle. To obtain the values of the two currents in

Fig. 363 at any particular instant, they are rotated together as is indicated in Figs. 364 and 365. The values will be equal to the lines *a a* and *b b*. In the same way the values of the three currents in a three-phase system are obtained for any instant as is illustrated in Figs. 367 and 368.

For the transmission of the currents of a two-phase system, three or four wires can be used. In the three-phase system, if the three currents are equal, three wires are sufficient, but if these currents are not equal a fourth wire is required to carry the surplus



Figs. 366, 367 and 368. Phase relations for three-phase system.

current as it may be called. When the three currents of a three-phase system are equal it is called a balance system, but if they are not equal the system is unbalanced. In Figs. 366 to 368 the three currents are drawn of equal length and it will be found that in every position in which the lines can be placed the sum of the two currents on one side of line *S S* will be just equal to the current on the other side, so that if the current is flowing away from the generator through one wire, it will divide up and return through the other two, and provide for each wire just the amount of current required. Thus in Fig. 366 the current flowing in *A* is zero, and the positive current in *B* is equal to the negative current in *C*. In Fig. 367 the two positive currents *a a* and *b b* in lines *A B*, are just equal to the one negative current in *C*, and this is also the case in Fig. 368.

Unbalanced three-phase currents are seldom used, but when they are, a fourth wire is run from the point where the three circuits *A B C* are joined, to a corresponding point at the generator end of the circuit, and then any excess or deficiency of current that is not provided for by the three regular circuit wires is taken through the fourth wire. The point where the three wires join, at the center of the circle, is called the neutral point, and the wire connecting then is the neutral wire. Two and three-phase systems are used almost exclusively for the transmission of power to great distances, and for this work only three wires are used.

Polyphase systems can be formed of any number of currents, but they would be of no practical value, owing to increased complications, and on that account are not used. In addition to the one, two and three-phase systems, explained in the foregoing, the only system that has been used to any extent is the "monocyclic," which was introduced by the General Electric Company. This system may be described as a sort of cross between the single phase and the polyphase systems. It consists of two currents, 90 degrees out of phase, just as in a two phase system, but instead of the two currents being equal, one of them is four times the strength of the other. The armature coils of the generators that furnish these currents are so connected with each other that the two currents, as fed into the line wires, constitute an unbalanced three-phase system. This arrangement of the generator coils will be found more fully explained in the section on "Alternating Current Generator," and the object of the "monocyclic" system will be found explained in the section on "Transmission Systems."

Inductive action in alternating current circuits. — In Fig. 369 let *G* represent an alternating current generator that impels an alternating current through the circuit *A A*. This current as already explained will develop a magnetic flux around the wire such as is indicated at *C D*. This flux will develop a self-induc-

tive e.m.f. in the circuit and thus retard the current, so that the actual amount of current flowing will be less than it would be in a continuous current circuit acted upon by an impressed e.m.f. of the same magnitude. As will be noticed, the direction of the flux at *C* and *D* is such that they oppose each other, that is the lines *C* and *D* flow through the space between the two sides of the loop *A A* in opposite directions, and on that account the lines *C* can only extend to the center of the space, while lines *D* will occupy the upper half. This being the case it is evident that if the circuit wires are brought closer together as indicated by the lines *B B*, the magnitude of the magnetic flux that will surround each wire will be correspondingly reduced as is indicated by the lines *a a*. The self-inductive e.m.f. developed in the circuit will be proportional to the magnitude of the flux that surrounds the wire, hence the nearer the two sides are brought to each other the less the self-induction, and if the two wires could be placed side by side the inductive effect would be practically nothing. From this it

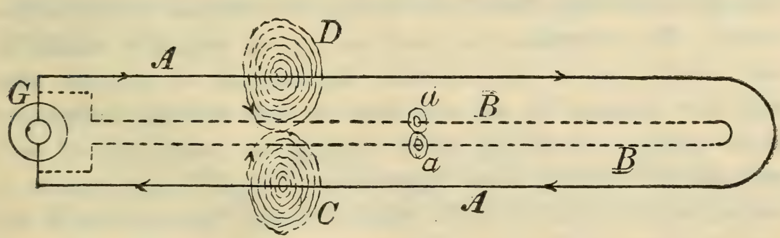


Fig. 369. Inductive action in alternating current circuits.

will be seen that if an alternating current is transmitted to a distance the nearer the line wires to each other the smaller the self-induction developed in them.

In an alternating current circuit the self-induction developed in every portion is not the same, and the total effect is equal to the sum of the several effects. For example in Fig. 370 let *A A A* represent a circuit that is fed by a generator at *G*. The self-

induction on the line A will be small, specially if the wires are placed near each other. If a number of incandescent lamps are connected at C the self-induction of these will be practically nothing. If at B we place some kind of device that is provided with wire in the form of coils, then at this point a large self-induction will be developed, for then the magnetic flux from each turn of wire in the coil will be able to cut

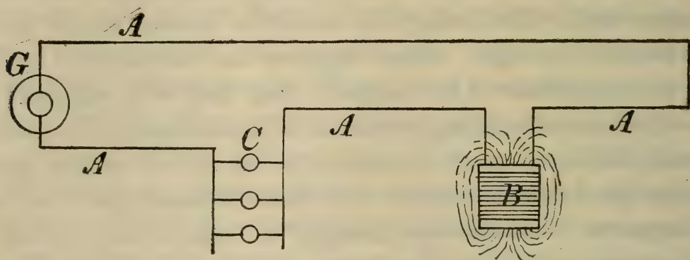


Fig. 370. Illustrating alternating current circuit.

through many other turns, and thus greatly increase the inductive action. To determine the total amount of inductive action in this circuit, so as to ascertain the amount of current that will flow through it, we will have to find the total impedance of the circuit, and this we do by finding the impedance of each part and then adding these impedances, but all this operation is carried out not in the way in which we add figures, but in the manner shown in Fig. 359. The diagram Fig. 371 illustrates the operation. By actual measurement we can find the resistance of the line A in ohms and we can mark it down on the diagram as $o\ a$. By calculation, we find the reactance of line A and mark it down as $a\ a'$, thus we obtain the impedance of $o\ a'$ of the line. Next, we find the resistance of the lamps C which we mark down at $a'\ b$, and from b draw $b\ b'$ equal to the reactance of the lamps, thus obtaining the impedance $a'\ b'$, of the lamps. We now draw $b'\ c$ equal to the resistance of B and $c\ c'$ equal to the reactance of

B and thereby obtain the impedance *b′ c′* of *B*. We now join *o*

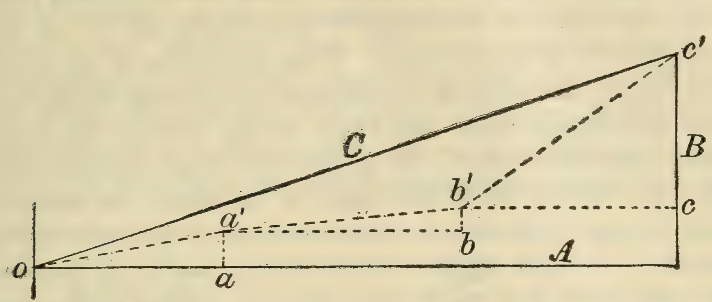


Fig. 371. Determining total inductive action.

with *c′* and obtain the line *C* which is the total impedance of the circuit, and line *B*, which is the total reactance, while line *A* is the total resistance. A glance at the diagram will show that the total impedance *C* is less than the sum *o a′ a′ b′* and *b′ c′* if these were added in the ordinary way, so that the total impedance of a circuit can be less than the direct sum of the impedances of its several parts.

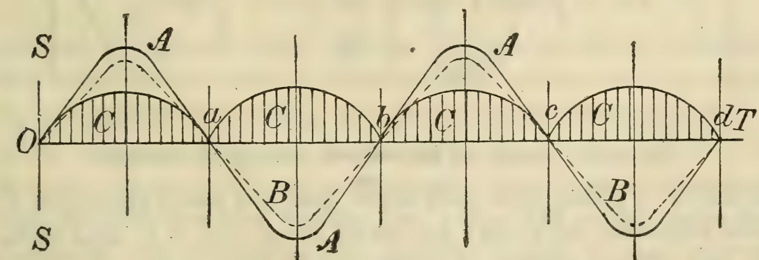


Fig. 372. Showing e.m.f. and current in phase.

The angle of lag between the current and impressed e.m.f. in an alternating circuit plays a very important part in determining the actual amount of energy that is transmitted. In a continuous current circuit the energy is always equal to the

product of the volts by the amperes but in an alternating circuit it may be equal to this product and it may not be as much as one per cent of this product. What proportion of the product of the volts by the amperes will represent the actual energy transmitted will depend upon the angle of lag between the current and the impressed e.m.f., the greater this angle the less the energy. The way in which the angle of lag affects the amount of energy flowing in the circuit can be made clear by means of Figs. 372 to 374. In these figures, curve *A* represents the impressed e.m.f. and curve *B* is the current, while the shaded curves represent the energy. In Fig. 372 the impressed e.m.f. and the current

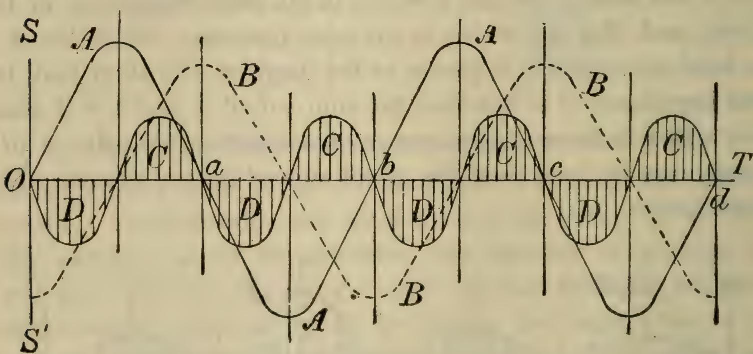


Fig. 373. Angle of lag affects energy in circuit.

are shown in phase with each other, and as a result the curves *C*, which represent the energy are drawn above line *O T*, thus showing that all the energy is positive, and it is equal to the direct product of the volts by the amperes. In Fig. 373 the current and impressed e.m.f. are drawn out of phase 90 degrees. Starting from *O*, the e.m.f. is positive while the current is negative, curve *B* being below line *O T*. This means that the current and e.m.f. act against each other hence the energy represented is negative. After the first quarter of a period, the current becomes positive

and then the energy is positive. Thus for the first half period we have two energy curves, D negative, and C positive, both of these are equal and, therefore, just offset each other, so that the net energy flowing in the circuit during this time is zero. As will be seen, during the following half periods, the same operation is repeated, so that the actual result is that energy is put into the circuit during one quarter period, and during the next quarter it is taken

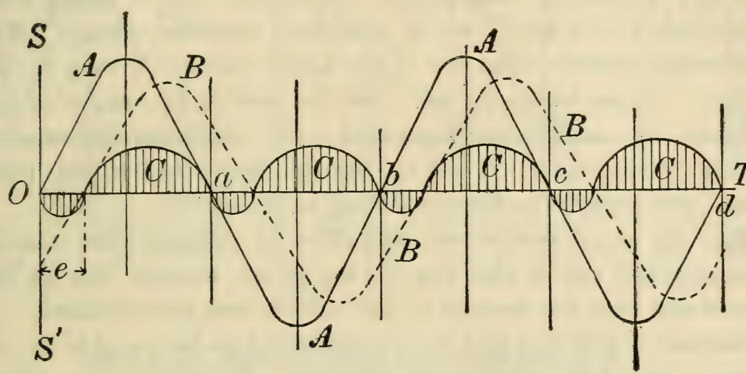


Fig. 374. Energy in circuit affected by lag.

out, and the actual energy flowing through the circuit is nothing. The action is the same as when a swing is set in motion, during the first half of each swing energy is accumulated by the descent of the weight, but during the next half it is all absorbed in lifting the same weight, and unless we supply from outside enough energy to overcome the friction the swing will soon come to a standstill. In an alternating current circuit, if the impressed e.m.f. and the current were out of phase 90 degrees no energy would be introduced into the circuit, hence, no current at all could flow, but if the angle is a trifle less than 90, say 89, a sufficient amount of energy can be put into the circuit to overcome the resistance loss, and then a strong current will sway back and forth that is not capable of doing any work. A current of this

kind is called a wattless current as it carries no energy. The reason why it carries no energy is that the self-induction very nearly balances the impressed e.m.f. so that the effective e.m.f. is very small, in fact it is just enough to force the current against the resistance of the circuit.

In Fig. 374 the current and impressed e.m.f. are shown out of phase by an angle of 45 degrees, and as will be seen the shaded curves *C* which represent positive energy, are much larger than those below line *O T*, which represent negative energy. The difference between these two is the actual energy flowing in the circuit. It can be clearly seen that the smaller the angle of lag between the current and impressed e.m.f. the larger the shaded curves above line *O T* and the smaller those below the line; hence, the greater the energy flowing in the circuit.

By the use of condensers, the effect of self-induction can be counteracted, and in that way the lag of the current can be reduced and thus the energy in the circuit can be increased. A condenser is a device that is so constructed as to be able to receive a very large electriostatic charge. To explain the nature of electrostatic charges so that they may be understood we may say that bodies arranged so as to hold a charge will carry this charge upon their surface. Thus we can picture to the mind's eye the charge as flowing over the surface until it completely covers it. When a condenser is used in an alternating current circuit, it is charged and discharged each time the current alternates, and the time relation of the charging and discharging currents is such as to be directly opposite to the current that would flow under the effect of the self-induction, or, to put it in another way, the e.m.f of the condenser current is 180 degrees out of phase with the self-induction. Now, by properly proportioning the condenser it can be made to just balance the self-induction, and then we get the relations illustrated in Fig. 375 in which curve *B* represents the self-induction, curve *C* the condenser e.m.f. which

is directly opposite and of equal magnitude. Curve *A* represents the impressed e.m.f. as well as the current, both being in phase with each other.

The general principle of construction of a condenser is illustrated in Fig. 376, in which the plates *A B* represent the condenser,

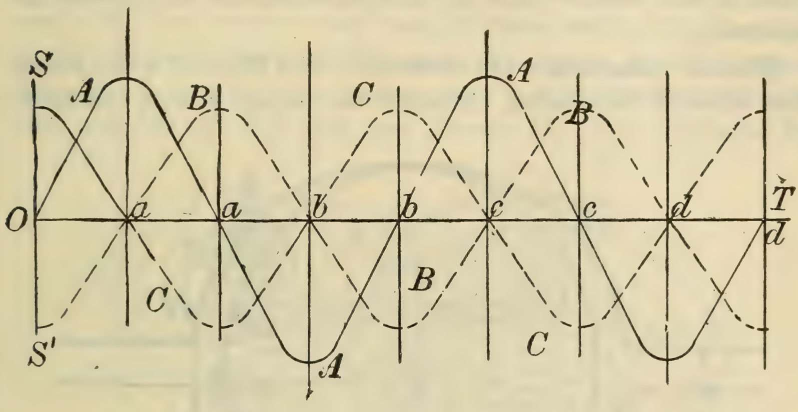


Fig. 375. Self-induction and condenser e.m.f.

and *G* the generator that provides the current, the connecting wires being *S S*. A device of this kind, if placed in a continuous

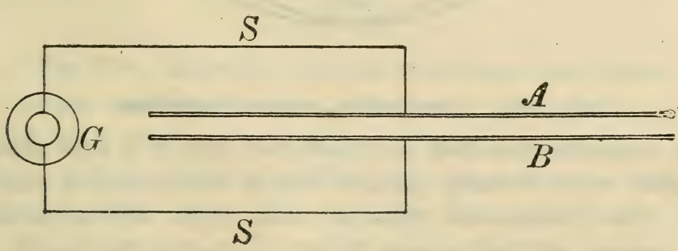


Fig. 376. Principle of the condenser.

current circuit, will simply prevent the flow of current; but when connected in an alternating current circuit, if of the proper proportions, will act as if it did not break the circuit. This is because

the large surfaces on the plates *A B* act as reservoirs and accumulate all the current that flows into them during the short time each impulse lasts. When the current reverses, the charge in the condenser runs out together with the generator current. We can thus consider that if a positive impulse of the current fills plate *A* and empties plate *B*, a negative impulse will reverse the operation.

Mutual induction. — In connection with Fig. 351 it was shown that when an alternating current flows through a wire, the alter-

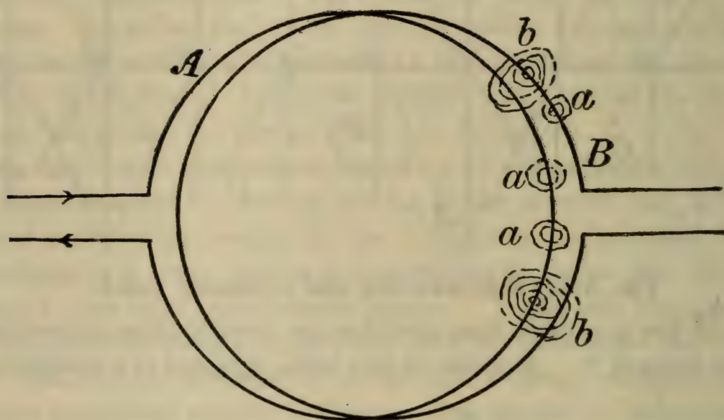


Fig. 377. Illustrating mutual induction.

nating magnetic flux that surrounds the wire, if it cuts through any other wires running parallel with it will induce e.m.fs. in them. The direction and phase of these e.m.fs. will be the same as that of the self-induction in the wire carrying the current. If we have two wires running parallel with each other and alternating currents flow through, then the action of wire No. 1 upon wire No. 2 will be the same as that of No. 2 upon No. 1. This action is called mutual induction, and it is made use of in the

construction of an apparatus used for transforming alternating currents which is commonly called a transformer.

By the aid of Fig. 377 the principles of mutual induction can be made quite clear. In this diagram suppose that the circle *A* represents one wire through which an alternating current is flowing, and circle *B* represents another wire carrying an alternating current. If these two wires are some distance apart, it is clear that a considerable portion of the magnetic flux of *A* will not cut through *B*, and in like manner that a considerable portion of the flux of *B* will not cut though *A*, as is indicated by

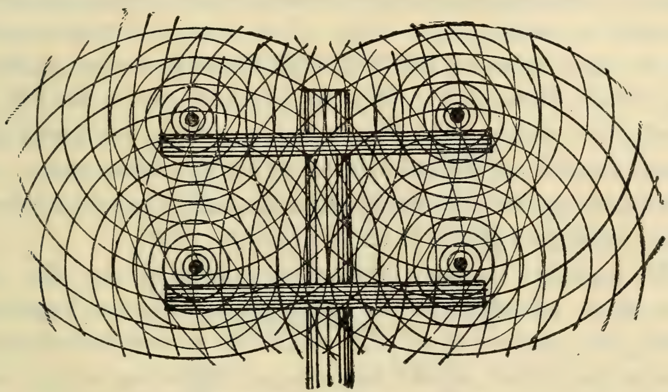


Fig. 378. Inductive effect of wires upon each other.

the dotted circles at *a a a*. In any case, however, some of the flux of one wire will cut through the other. From this it follows that the effect of the current in each wire upon the other wire will be less than that upon itself, but the closer the wires are to each other the nearer equal the effects will be. When it is desired to avoid the effects of mutual induction as far as possible the wires must be separated to the greatest distance, and when we desire to make the mutual inductive effect the greatest, we must bring the wires as close

together as possible. The inductive effect of wires upon each other in some cases produces very objectionable results, for example when telephone wires are run side by side for any distance the inductive action of one wire upon the other serves to render the conversation indistinct. Why this is so it can be appreciated at once from an inspection of Fig. 378, which shows a pole carrying four wires. Telephone currents are not alternating but they pulsate and thus produce the same effect as if they were alternating. In Fig. 378 the circles drawn around each one of the wires as will be seen cut through all the other wires. If the two upper wires belong to one circuit and the two lower ones to another, then if one set of wires are crossed at every three or four poles so that the wire running on the right side for a certain distance will then be changed over to the left side, the inductive actions will be counteracted to a very great extent and this method is followed in stringing telephone wires. It is also used in regular alternating current circuits when interference between different circuits is to be avoided.

With regards to the two wires belonging to the same circuit, it is advantageous to string them as close together as possible, for in this case, the effect of mutual induction is to neutralize the effect of self-induction. Referring to Fig. 369 it can be seen at once that if the magnetic flux at C develops a self-induction in lower A toward the right, it will develop an induction in upper A also towards the right, but with reference to the wire itself this induction will be just opposite to that in the lower side so that the two will counteract each other. Thus to reduce the reactance of the line, the two sides of the circuit must be placed as near together as is practicable.

Transformers. — A transformer is an apparatus in which the principle of mutual induction is utilized for the purpose of generating a second current by the inductive action of a primary current. Referring to Fig. 377 it can be seen that if wire B is

closed upon itself the e.m.f. induced in it by the magnetic flux issuing from A will cause a current to flow and then this current, which is brought into existence by the inductive action of the current in A, will in turn develop a magnetic flux that will react upon wire A in precisely the same manner as if the current were not induced in B, but it came from an independent source. In a transformer, the wire is wound in the form of compact coils, and one of these coils, which is called the primary, is connected with an alternating current circuit. The current flowing through this coil induces a current in the other coil which is called the secondary. The general construction of a transformer can be under-

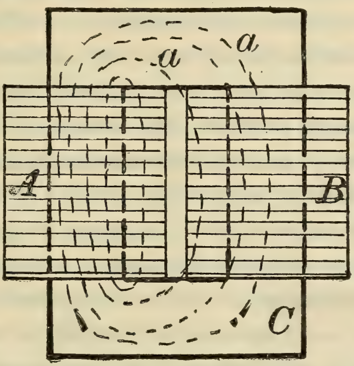


Fig. 379. Principle of the transformer.

stood from Fig. 379. An iron core C is provided upon which are wound two coils marked A and B. The coil A which is the primary, is connected with an alternating current circuit, and thus the iron core C is strongly magnetized. The presence of the iron core C serves to greatly increase the magnetic flux but does not in any way interfere with its alternating properties, so that it increases and decreases and changes its direction in precisely the same manner as the flux that surrounds a single wire. The flux de-

veloped by A, swells out as indicated by the lines a a a and cuts through the side of the secondary coil B. If the circuit through this coil is close an alternating current will be generated in it, and this current will develop a magnetic flux that will swell out and cut the side of the primary coil A. The e.m.f. induced in A by the flux of B will be in opposition to the self-induction developed by its own flux, hence, if the circuit through B is open, the current flowing through A will be small because the self-induction will counteract the impressed e.m.f. so as to leave but a small effective e.m.f. As soon as the circuit through B is closed, the inductive action of this coil upon A will offset to a certain extent the self-induction and thus assist the impressed e.m.f. in forcing more current through A. The more the current through B is increased, the stronger its action upon A and as a result the more the self-induction of A will be neutralized and the stronger the primary current will become. This action, which occurs in a perfectly natural manner, serves to make the transformer a self-regulating apparatus, so that if a strong current is required in the secondary circuit, a strong current passes through the primary so as to furnish the energy necessary to develop the strong secondary current. If no current is drawn from the secondary, the primary current is reduced to nearly nothing.

To explain fully the action in a transformer would require a rather lengthy discussion of the principles involved, but the action, in a general way, can be made clear without going deeply into the theory. In explaining the phase relation of the current, the self-induction and the impressed e.m.fs. in connection with Fig. 357 it was shown that theangle between the self-induction and the current is 90 degrees, and that the angle between the current and the impressed e.m.f. can be anything from zero up to nearly 90 degrees. If the current is passed through transformers or other inductive devices, the current will lag considerably. Suppose it lags 10 degrees, then the total angle between the im-

pressed e.m.f. and the self-induction will be 100 degrees. Now in a transformer the e.m.f. induced in the secondary coil is in phase with the self-induction in the primary coil, hence, with the above angles it would be 100 degrees behind the impressed e.m.f. in the primary coil. Now if the secondary current lags as much as the primary, it will be 110 degrees behind the primary impressed e.m.f. and the magnetic flux developed by this current will induce an e.m.f. in the primary coil 90 degrees behind itself or 200 degrees behind the primary impressed e.m.f. This e.m.f. induced in the primary coil by the action of the secondary current not only counteracts the self-induction in the primary coil, but in addition changes the phase relation between the primary current and its impressed e.m.f., making the angle smaller. This change in the phase relation between the current and impressed e.m.f. results, in turn, in a change of the phase relation of the secondary current, and this change in the phase of the secondary makes a corresponding change in the phase of the primary. If we were to trace up the action back and forth from primary to secondary currents we would finally arrive at the true phase relation of the currents and e.m.fs. in both circuits but this is a complicated and unnecessary process of reasoning. We can easily see that the current induced in the secondary coil will have a certain phase relation with respect to the primary current, and we can further see that the combined magnetizing effect of the two currents, the primary and secondary, is the same as that of a single current having a phase intermediate between the phases of these two. Following this course of reasoning we have only one inductive action to deal with and this is in such a phase relation that as it increases it decreases the self-inductive e.m.f. in the primary and thus permits more current to pass through this coil, and this increase in current in the primary causes a corresponding increase in the secondary current. When the secondary current is very small the self-induction in the

primary is very great and as a result the lag of the primary current is increased and its strength is decreased. As the secondary current increases, the self-induction in the primary decreases, and the lag of the primary current reduces while the current strength increases. The strength of the secondary current is varied by varying the resistance in the secondary circuit; if this resistance is reduced the current is increased.

To make a transformer as perfect as possible it is necessary to place the primary and secondary coils in such a position that the mutual induction between them may be the greatest possible, that is so that all the magnetic flux developed by the primary coil may cut through the secondary and all the flux of the secondary may cut through the primary. If the coils are arranged as in Fig. 379 it can be seen at once that all the flux of *A* will not cut through *B* and in like manner all the flux of *B* will not cut through *A*. It is not possible to arrange the coils so that all the flux of one coil will pass through all the turns of wire on the other coil, but this condition can be very nearly realized. If one-half of coil *A* is wound on each side of the core *C* and then the *B* coil is wound in two parts directly over the *A* coils the chance for the flux of one coil to not pass through the other coil will be greatly reduced.

The flux that does not pass through the opposite coil is called a leakage flux, thus in Fig. 379 the lines *a* that pass through coil *A* but not through *B* constitute the leakage from coil *A* and in like manner the flux of coil *B* that does not pass through *A* is the leakage of *B*. The leakage flux represents just so much magnetism thrown away, hence the effort of the designer is to arrange the coils so as to reduce it to the smallest amount possible. If the two coils were wound together, that is, if we took the wires and wound them side by side forming a single coil, the leakage would be practically nothing, but this construction cannot be used as with it there would be great danger of the insulation between

the coils giving away, and this would destroy the transformer. This form of winding can be approximated to by winding each coil in many sections and placing these in sandwich fashion upon

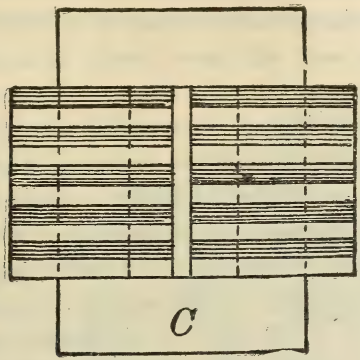


Fig. 380. Position of transformer coils on iron core.

the iron core as is shown in Fig. 380 in which the sections forming one coil are shaded, and those of the other coil are not. This is the construction that is followed generally in large transformers. In the majority of designs, however, the primary and secondary coils are wound one over the other.

Transformers are used for the purpose of changing the voltage of the current. The name transformer is misleading, as it creates the impression that the device transforms the current, when as shown in the foregoing it does nothing of the kind, it simply generates a secondary current, which is in no way connected with the primary. When electric energy is transmitted to a considerable distance by means of alternating currents, the voltage used is much higher than is required for the operation of lamps or motors, hence, at the receiving end of the line this current is passed through transformers and secondary currents are generated in these that are of the voltage desired. The voltage

of the secondary current is controlled by the number of turns of wire placed upon the secondary coils. Roughly speaking, if the primary coil has ten times as many turns as the secondary the voltage of the secondary current will be one-tenth of that of the primary. If the primary voltage is 2000 and the secondary is 100 the primary coil will have twenty times as many turns of wire as the secondary.

Transformers that deliver a secondary current of lower voltage than the primary are called lowering transformers, while those that deliver a secondary of higher voltage are called raising transformers. For distributing current to consumers, lowering transformers are used. But in long distance transmission plants, where the current in the transmission line has an e.m.f. of anywhere from 10,000 to 30,000 volts, raising transformers are used at the power house, and these take the current from the generators, which may be of 1,000 or 2,000 volts and deliver to the line a secondary current of 10,000 or more volts.

Transformers cannot be used with continuous currents for the simple reason that as these currents do not fluctuate the magnetic flux developed by them remains stationary and, therefore, there is no inductive action.

A medium size transformer is shown in Fig. 381. The complete transformer is seen at the right side of the illustration. In the center is shown the lower part of the iron core, with the wire removed from one leg, this wire being shown on the left. The iron plates at the bottom of the figure form the upper part of the iron core.

The iron core of a transformer is built up out of sheet iron. It could not be made a solid mass, for, if it were, secondary currents would be induced in it, and thus the energy in the primary current would be used up in developing useless currents in the iron core. The sheet iron laminations are insulated from each other, so as to prevent the development of currents in the core.

As can be seen from the illustration, the wire wound on each leg of the core belongs in part to the primary and in part to the secondary circuit. If the primary wire is proportioned so that it is proper for a 1,000-volt current when the parts on the two legs are connected in series, then it can be made proper for 500 volts

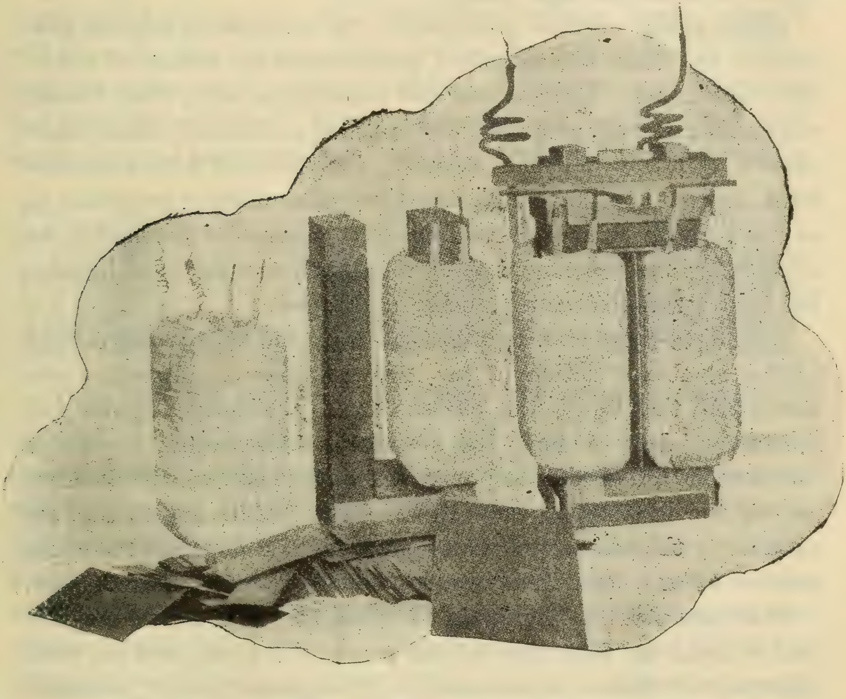


Fig. 381. Showing outer covering of transformer.

by connecting the two parts in parallel. If the secondary coils will develop a voltage of 100 when both parts are connected in series, they will develop 50 volts if both parts are connected in parallel, but in this case the current will be doubled.

The transformer as shown to the right in Fig. 381, is complete, but for the purpose of protecting the wire an outer casing is pro-

vided. For high voltage transformers, this casing is made water tight and is filled with oil so as to improve the insulation of the apparatus. Very large transformers are provided with means for cooling them. In some, air is forced through the coils and iron core. In others, coils of pipe are placed within the casing and water circulates through these.

Alternating current generators. In alternating current generators the field is magnetized permanently by means of a continuous current. This current is obtained, generally, from a small continuous current generator that is called an exciter. Some alternators, as a rule of small capacity, are provided with a commutator to rectify a portion of the current the machine generates so as to provide a continuous current to magnetize the field. An alternating current cannot be used to magnetize the field because the field magnetism must remain unchanged.

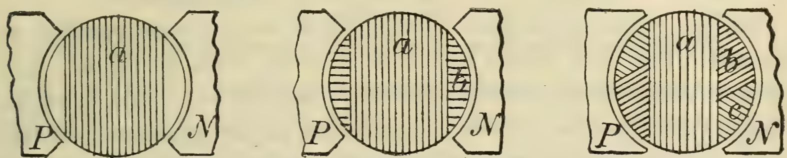
Alternators are also arranged so that the field is magnetized by the combined action of the two continuous currents above mentioned, that is, by the current from a separate exciter and the current derived from the armature. Alternators excited in this manner are called compound machines and are the counterpart of the continuous current generator. Alternators that are excited by the current from a separate exciter alone are the counterpart of the plain shunt wound continuous current generator.

There are several other ways in which the field can be magnetized to make an alternator of the compound type, and the most important of these will be found fully explained under the heading of "Compensated Generators."

The object of compound winding in alternators is the same as in continuous current generators, that is, to keep the voltage constant and not allow it to drop as the current strength increases. Large alternators used in central stations are always of the compound type.

The way in which alternating current generators act can be

understood from the diagrams Figs. 382 to 386. In Fig. 382 *P* and *N* represent the poles of the field magnet of a two-pole machine. The armature is provided with a single coil of wire marked *a*. When this coil is in the position shown, no e.m.f. will be induced in it, but as it begins to rotate from this position an e.m.f. will begin to be induced, and this will increase in magnitude until one-quarter of a revolution has been made, when it will be at the maximum value. During the next quarter revolution the e.m.f. will gradually reduce, becoming zero when the half turn is completed. During the next half turn the e.m.f. will again rise to a maximum and fall to zero, but it will be oppositely



Figs. 382, 383 and 384. Generating current in alternator.

directed, so that if during the first half turn the e.m.f. is positive, during the next half it will be negative, and this operation will be repeated for each revolution of the armature. Thus it will be seen that if the armature revolves ten times in a second, the frequency of the current generated will be ten, and in any case the frequency will be equal to the number of revolutions the armature makes in a second. This is true for a two-pole machine, if the generator has four poles the frequency of the current will be equal to twice the number of revolutions per second and for any greater number of poles the frequency will be equal to the number of revolutions of the armature per second multiplied by half the number of poles. Alternating current generators are always made with a large number of poles so that the frequency required may be obtained without running the armature at too great a speed.

The diagram Fig. 382 illustrates a simple alternating current generator, or what is called a single-phase generator. A single-phase machine is one that has one coil on the armature for each pair of poles in the fields and generates one alternating current.

Fig. 383 illustrates diagrammatically a two-phase generator. A two-phase generator is an alternating current generator that generates two alternating currents that are out of phase with each other by one-quarter of a period, that is, by 90 degrees. Such a generator is provided with two coils or sets of coils for each pair of poles and these are placed at right angles to each other in a two-pole machine, and so that the sides of one set come opposite the centers of the other set, in multipolar machines.

In Fig. 383 it will be seen that coil *a* is in the same position as the coil in Fig. 382, hence no e.m.f. is being induced in it. Coil *b*, however, is in the position in which the induced e.m.f. is of the maximum value, thus it will be seen that as the armature revolves the e.m.f. in one coil will rise toward the maximum while that in the other coil will be decreasing toward zero.

Fig. 384 illustrates a three-phase generator. A three-phase generator is a machine that generates three alternating currents that are out of phase with each other by an angle of 120 degrees, or one-third of a period. Such a machine has three coils or sets of coils for each pair of field poles.

In Fig. 384 it will be seen that coil *a* is in the position in which no e.m.f. is generated, and if we assume that the armature is revolving in the direction of the hands of a clock, then the e.m.f. induced in coil *b* is very near the maximum value, but is still increasing, and will become the maximum when the coil reaches the horizontal position. In coil *c* the e.m.f. has passed the maximum and is reducing toward zero, which value it will reach when the coil reaches the vertical position, or the position in which *a* now is.

If an alternator is of the multipolar type the coils will be dis-

posed in the manner shown in Fig. 385. If it is a single-phase machine it will have one set of coils only, those marked *A*. If it is a two-phase generator it will have two sets of coils, the addi-

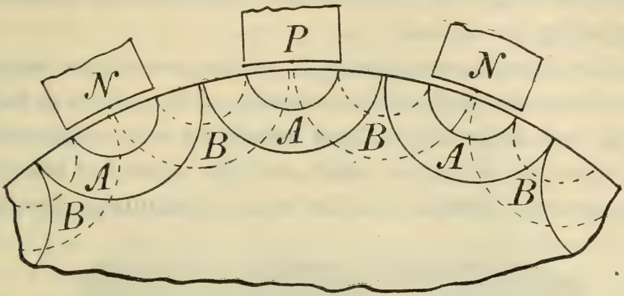


Fig. 385. Arrangement of coils in multipolar alternator.

tional set being placed in the position shown in broken lines and marked *B*. In this construction the machine appears to have as many *A* coils as there are poles and the same number of *B* coils, which is in contradiction to the statement made above that a single-phase machine has one coil for each pair of poles. The truth,

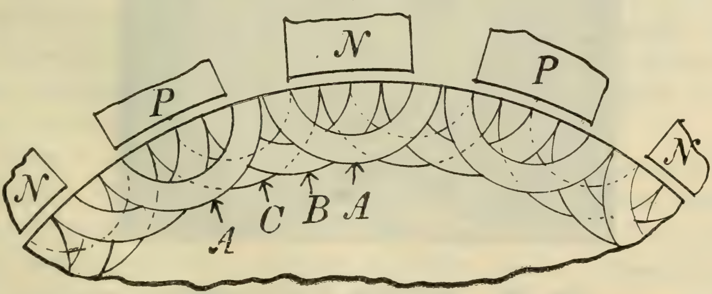


Fig. 386. Arrangement of coils in three-phase generator.

however, is that each coil in Fig. 385 is virtually one-half of a coil. Fig. 386 shows the way in which the coils are arranged in a three-phase generator of the multipolar type, the three sets of coils being marked *A B C*. In monocyclic generators the coils

are arranged as in Fig. 385, but they differ from the two-phase winding in that the *B* coils are one-quarter the size of the *A* coils. In actual generators the armature coils are seldom given the form shown in these diagrams, but whatever the form may be the principle of winding is the same.

In an alternator the armature coils forming one set are connected in series with each other, and the entering end of the first coil and the leaving end of the last coil are connected with collector rings mounted upon the armature shaft, and the current is taken from these by means of brushes similar to the commutator brushes of

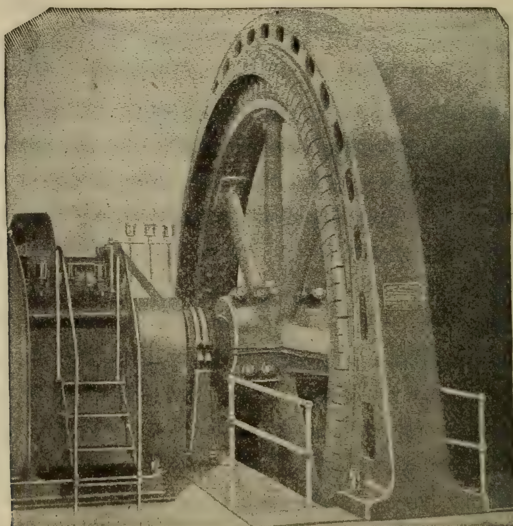


Fig. 387. Revolving field alternator.

continuous current machines. In monocyclic generators one end of the *B* set of coils is connected with the middle point of the *A* set, and the three remaining ends are connected with collector rings. This is the arrangement with generators of what is known as the revolving armature type, which is the

one illustrated in Fig. 382 to 386. There is another type in which the outer part, which is stationary, is the armature, and the revolving part is the field. Machines of this kind are called revolving field alternators. The principle of operation is the same in both types, but the revolving field type has the advantage that, as the armature is stationary, no collector rings and brushes are required to take off the current. All that is necessary is to provide binding posts to which the ends of the armature coils are connected, and from these the external circuit wires are run off.

A revolving field alternator is shown in Fig. 387. In machines of this type, the field magnetizing coils are mounted on the periphery of the revolving part, hence the current that traverses them must pass through collector rings mounted upon the shaft. These rings are clearly shown in the illustration, the collector brushes being held, insulated from each other, by the stand located in front of the rings. Thus it will be seen that this type of machine requires collector rings, just the same as the revolving armature type, but the difference between the two is that in the latter the whole armature current passes through the collector rings, and on that account they must be made very large, while in the revolving field machines they can be made small, as only the field current passes through them, and this is only from one to two per cent of the armature current.

There is still another type of alternating current generator in which the wire on the field as well as the armature is held stationary. Such machines are called inductor generators. The revolving portion of such generators is simply a mass of iron formed like a very large pinion with correspondingly large teeth. When this part revolves the ends of the teeth sweep over the armature coils, running as close to them as they can without touching. The magnetic flux developed by the field coil issues from the ends of the teeth and cuts through the armature coils thus inducing e.m.fs. in them. It will be seen that the difference between this type of

generators and the revolving armature type is that instead of revolving the armature coils through the stationary field flux, the latter is revolved and the armature coils are held stationary. The

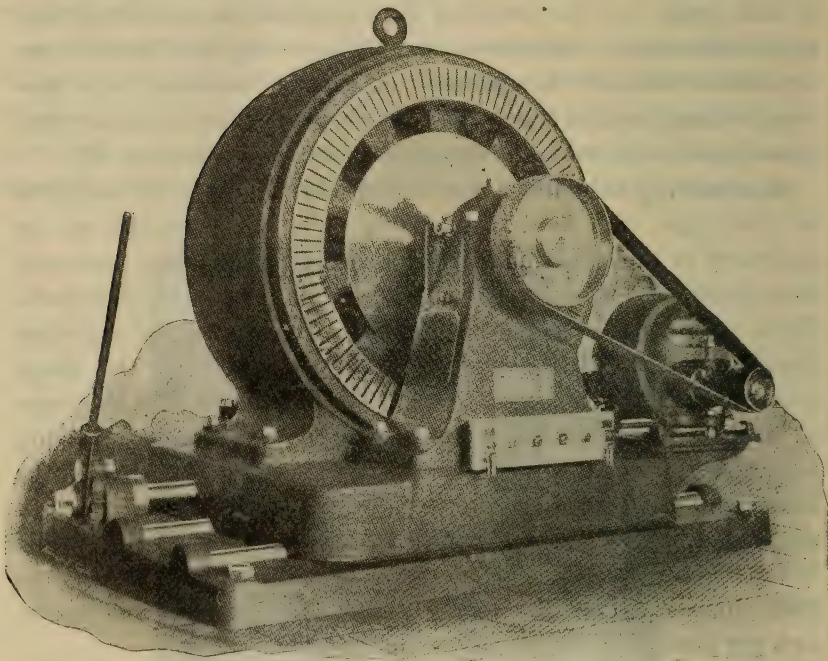


Fig. 388. Inductor alternator.

difference between the inductor generator and the revolving field type is that in the latter the field is magnetized by a number of coils and these are rotated together with the field poles, while in the inductor machine there is a single field magnetizing coil and this remains stationary, the part that revolves being what might be called the poles.

An inductor alternator is shown in Fig. 388. The small machine mounted on the right side of the base is the exciter that

furnishes the field magnetizing current. The outer casing of the machine holds a ring built up of sheet iron laminations, which constitutes the armature and supports the armature coils. The large teeth, or polar projections, which are well shown in the illustration, are carried by the revolving part, and when rotating cause the magnetic flux to sweep over the armature coils. The field coil is placed back of these polar projections.

Alternating current generators are run singly, or they may be connected in parallel, but they cannot be run in series. If an attempt is made to run them in series, one of the machines will act as a motor and will be driven by the current generated by the other. When alternators are connected in parallel it is necessary that they run at exactly the same velocity, if they are identical in construction. If the generators are not of the same construction then their velocities will depend upon the number of poles each one has. Machines of different size and even design, can be connected in parallel, providing the frequency of the currents they generate are the same. To make the frequency the same it is necessary that the velocity of each machine multiplied by the number of poles it has be equal to the same number. Thus if one machine has twice as many poles as the other, it must run at one-half the velocity. The velocity of alternators connected in parallel must be equal, absolutely, and not practically so; that is, if two machines are alike, and one runs at 1000 revolutions per minute, the other must run at 1000 and it cannot run at 999 or 1001. Since such extreme accuracy in speed is necessary it might be inferred that it is practically impossible to run alternators in parallel unless their shafts are coupled together, or they are connected through spur gearing with the same driving shaft. As a matter of fact, however, alternators can be run in parallel even if one is driven by a steam engine and the other by a water wheel, and they may be side by side or several miles apart. The reason why this is the case is that when the machines are in oper-

ation, the current holds them in step. If several generators are feeding into the same circuit, and one machine tends to lag behind the others, its current reduces and thus the speed increases as less power is required to drive it. If the tendency to lag increases, the machine begins to act as a motor, and is driven by the current from the other machines.

While it is possible to run alternators in parallel under almost any conditions providing they are speeded so as to generate currents of the same frequency and nearly the same voltage, entirely satisfactory results cannot be obtained unless the angular motion is uniform, that is, unless the velocity of rotation is the same at all points of the revolution. If a steam engine has a light fly-wheel the velocity of the shaft will not be the same at all points of the revolution, but will be the slowest when the crank is passing the center, and the fastest when at half stroke. This fact is clearly shown by the irregular motion of the paddle-wheels of river boats driven by a single engine.

If two alternators are driven by two engines whose rotative motion is not uniform and the engines are so timed that one is on the center when the other is at half-stroke, then the action of the two alternators will be irregular, for when one machine is rotating at the highest velocity the other will be rotating at the lowest. This uneven action of the alternators may be compared with the work of two horses hitched to a wagon and pulling unevenly. If both horses pull together all the time the whiffle-tree will remain straight and the wagon will be drawn along smoothly; but as soon as the horses begin to pull unevenly the whiffle-tree will be jerked back and forth and the motion of the wagon will be irregular. In this case the horses soon tire out because they work against each other part of the time. The action between two alternators that do not rotate with uniform velocities is practically the same as that of two horses that do not work together; the machine that runs ahead not only sends a

current into the main circuit, but in addition backs up a current through the other generator, thus wasting energy by causing a strong current to flow back and forth between the two machines. To overcome this difficulty engines made to drive alternators are provided with extra heavy flywheels, so that the momentum may be sufficient to keep the speed up to the normal point while the crank is passing the center.

With small alternators that have only a few poles and are driven by high-speed engines, the affect of irregular motion is not so great as in large machines having many poles, hence the large slow-speed engines used to drive alternators having a large number of poles, must be provided with excessively large flywheels to run in a satisfactory manner.

The reason why alternators with a large number of poles require greater regularity in motion to give satisfactory results, can be easily understood. Suppose we have a pair of two-pole machines driven by engines whose flywheels are 25 ft. in circumference. Suppose, further, that the irregularity in motion is such that each engine when running at the faster velocity, gets three inches ahead of the other. Then the advance in position will be one per cent, and consequently the currents of the two generators will run ahead and behind each other one per cent at each quarter of a revolution. Now, if these same two engines drive two twenty-pole alternators, then the irregularity in motion will be multiplied ten times, because one-tenth of a revolution will give one cycle of current, and the current of each machine will run ahead and fall behind the other ten per cent, instead of one per cent.

Starting alternators connected in parallel: — In starting continuous current generators that are connected in parallel all we have to do is to set one machine in operation and then after the second one is running up to full speed, we adjust its field regulator until the voltage is the same as that of the first machine, or one or two volts higher. We then throw the switch and connect

it with the switchboard. In starting alternators that are connected in parallel we have to do more than this, we must not only adjust the second machine so that its voltage is the same as that of the first, but we must bring it up to the proper speed and get its current in phase with that of the first generator before we connect it with the switchboard. To accomplish all this with certainty, devices are used that are called synchronizers, or phase indicators. These devices consist generally of a couple of small transformers one of which is connected with the circuit of each generator. The secondary wires of these transformers are connected with each other and one or two incandescent lamps are connected in this circuit. When the second machine is started up, as its speed is much lower than that of the generator already in operation the frequency of the secondary current of its transformer will be much lower than that of the first machine, and as a result the lamps in the circuit of the two transformers will flicker rapidly. As the second machine builds up its speed the flickering of the lamps will become slower. When the two generators are running at nearly the same speed the flickering will be replaced by rather long periods of darkness and light. During the periods when the lamps are lighted the current generated by one of the transformers is in such a direction as to act in series with the current of the other and thus draw the current through the lamp. When the lamps are dark it is because the currents of the two transformers are in opposition to each other and thus no current passes through the lamps. The second generator is connected with the switchboard during one of the periods of darkness or brightness, depending upon the way in which the transformers are connected. The second generator will not be running at exactly the proper speed when it is connected with the switchboard, but as soon as it is connected the currents of the two machines acting upon each other will at once draw the second machine into step with the first one, and they will continue to run in step even if the power

driving one of the machines should fail. In the latter case, the first machine would not only furnish current for the main circuit, but would in addition drive the second machine as a motor.

The way in which synchronizing lamps are connected in single or polyphase circuits is clearly illustrated in the diagram Fig. 389·

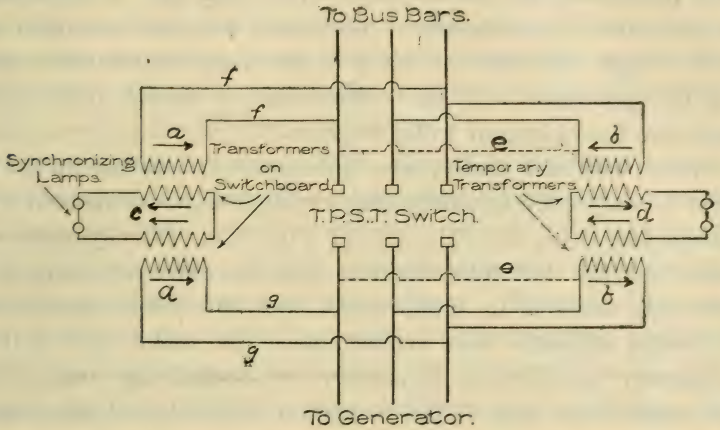


Fig. 389. Showing arrangement of synchronizing lamps.

The three upper lines are connected with the main bus-bars on the switchboard and the lower lines run to the generator that is to be synchronized. The left side of the diagram shows the connections for synchronizing a single-phase generator. In such a case, the middle wire running to the bus-bars and to the generator would not be used. The synchronizing transformers would have their primary coils connected with the side wires in the manner shown by lines *f f* and *g g*. When the generator current is in synchronism with that in the bus-bars, the primary currents in the two synchronizing transformers will flow in the direction of the arrows *a a*, and the secondary currents will be in the direction of arrows *c*, that is, in opposition to each other, so that no current will pass through the synchronizing lamps. If the connections of one of

the transformers are reversed, either in the primary or secondary, the two secondary currents will flow through the lamps in the same direction as indicated by the arrows *d* on the right side of the diagram. Thus it will be seen that the synchronizing lamps can be arranged so that they will light up when the generator current is in phase with the bus-bar current, or they may be arranged so as to be dark at this instant. Generally they are arranged so as to be bright when the current is in phase and the switch connecting the generator with the switchboard is closed at the instant when the lamps appear to be brighter.

When two and three-phase generators are started up the first time a temporary synchronizing arrangement is connected in the manner shown on the right side of Fig. 389. The synchronizing lamps on the left side will show that the current flowing in the two side wires is in synchronism, but this does not show that the other currents also synchronize. To make sure that the temporary transformer is properly connected the connections *e* are made first, and if the lamps on both sides of the diagram become dark and bright together, the connections are correct. The connections are then broken and are transferred to the middle wire; then when all the currents are synchronized, all the lights will light up together. Generally the internal connections of synchronizing transformers are properly made, and the correct connection of the terminal wires is clearly indicated so that mistakes in making connections are not very liable.

Compensating and compounding alternators.—Continuous current generators are provided with a compound field winding for the purpose of maintaining the voltage uniform as the armature current increases. Alternating current generators are compounded for the same purpose. If the field of an alternator is excited by a current derived from an exciter the voltage of the machine will drop as the strength of the current generated in the armature increases. A part of the drop is due to the fact

that the increased current absorbs more voltage in passing through the armature coils. The balance of the drop is produced by the reaction of the armature current upon the field. As the current of the exciter that magnetizes the field remains constant, the magnetization produced by it remains constant. The current flowing in the alternator armature acts to demagnetize the field, and, as its action increases as the strength increases it follows that the stronger the current becomes the weaker the field will be, and, as a result, the lower the voltage of the current generated in the alternator armature.

If a portion of the current of the alternator armature is rectified by being passed through a commutator and is used to assist the exciter current to magnetize the field then the field magnetism will increase as the armature current increases, because the action of the rectified current will increase. Thus by the compound action of the exciter current and the rectified armature current, the magnetism of the field of the alternator can be made to increase as the armature current increases, and in this way the voltage is increased so as to compensate for the greater drop of voltage on the armature coils, the result being that the voltage impressed upon the wire remains practically the same for all strengths of current.

The above results can be obtained providing the phase relation between the current and the impressed, or line e.m.f. does not change; but if the phase relation is continually changing such perfect regulation cannot be realized. The reason why changes in the phase of the current interfere with the regulation is that the same strength of armature current will produce different degrees of reaction on the field magnetism with different phase relations. If the lag of the current is increased the reaction upon the field will be increased, and in like manner a decrease in the lag will reduce the reaction upon the field. Several arrangements are used for obtaining field magnetizing currents that will com-

pensate for variations in the lag of the current as well as for variations in strength. Alternators provided with such arrangements are called "Compensated Generators." The way in which a field magnetizing current is obtained that will compensate for variations in lag as well as in current strength is by using a portion of the armature current to vary the strength of the current generated by an exciter, the exciter being provided with coils through which the current taken from the armature is passed. These coils are so disposed that their governing action upon the exciter is proportional to the lag of the current as well as its strength, hence the current that the exciter sends through the field coils of the alternator is at all times sufficient to compensate for variations in the strength and phase of the armature current.

If an alternator is single-phase, one commutator is sufficient to rectify the portion of the armature current and to magnetize the field. For a two-phase machine, two commutators are required and for a three-phase, three commutators. To obviate using two and three commutators in polyphase generators, transformers are employed, two transformers for two-phase and three transformers for three-phase. The recording currents of these transformers are combined into one, and this combined current is passed through a single commutator to be rectified. In some cases only one of the currents of a two or three-phase generator is rectified, but with most machines, if they are connected in parallel, care must be taken to have the circuits from which the rectified current is taken properly connected with each other; if not, one armature will short circuit the other. This is due to the fact that when alternators are run in parallel the rectified currents for the field coils are connected with each other through equalizer wires, in a manner similar to that used with continuous current generators.

The ordinary connections for two generators in parallel are shown in the diagram Fig. 390.

As will be seen, the field-magetizing currents derived from the commutators are connected with each other through the equalizer switches, hence, to avoid short circuiting the armature through the equalizer connections, if the commutator rectify one current only,

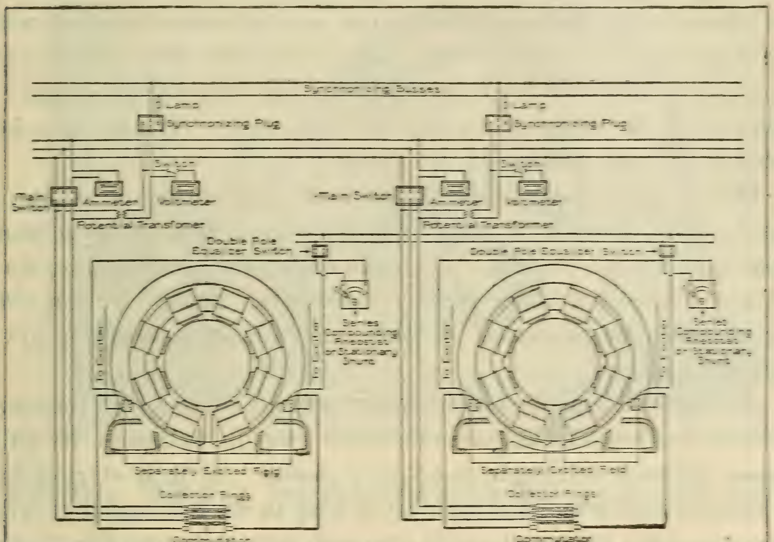


Fig. 390. Connections of composite field alternating generators for running in parallel.

the two rectified currents must be in phase with each other. The rheostats shown in each field circuit are for the purpose of adjusting the voltage of each generator independently.

The use of transformers to transform the portion of the armature current that is rectified is no objection against polyphase machines, because, even with single phases, the armature voltage is generally so high that a transformer is used so as to obtain a secondary current of low voltage to pass through the field coils.

Alternating current motors.—From the foregoing it can be understood that an alternating current generator can be used as a motor providing it is supplied with the same kind of currents.

that is, with a continuous current to magnetize the field, and with an alternating current for the armature. A single-phase alternator will run as a motor if connected in a single-phase circuit. Two-phase generators will act as two-phase motors, and three-phase generators will act as three-phase motors. With either one of these three types of machines a continuous current will be required to magnetize the field. Two and three-phase machines can be run with a single alternating current, by connecting one of the armature circuits only, or all the circuits may be used if they are connected in parallel.

When an alternator is used as a motor it is called a synchronous motor, because it runs in synchronism with the generator that supplies the current. A simple alternator (single-phase machine) becomes a single-phase synchronous motor, and a two or three-phase generator becomes a two or three-phase synchronous motor.

A single-phase synchronous motor will not start up of its own accord, but must be set in motion and run up to nearly its full speed before it will begin to act as a motor. If it is started up without a load when it comes rather near to its full speed it will give a sudden jump and swing into step with the current and then continue to run at this velocity. If it is started with a full load it will not fall into step with the current until its speed is very nearly up to the proper point. Synchronous motors are never started under load, they are always started light.

Two and three-phase synchronous motors can be started without outside assistance. Synchronous motors are generally provided with a self-starting motor, to set them in motion, or else they are arranged so as to be self-starting by being converted, in the act of starting, into some form of motor that is self-starting.

Fig. 391 shows a synchronous motor of large size provided with an induction motor of much smaller capacity to start it.

This motor is of the revolving field type, and, as will be seen, is precisely the same as the same type of generator.

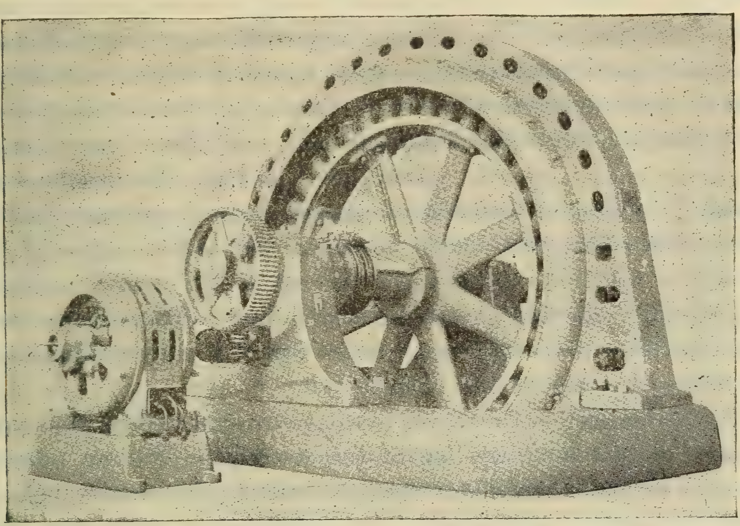


Fig. 391. 1000 h. p. two-phase revolving field synchronons motor.

Owing to the fact that synchronous motors are not self-starting, they are generally used only where large power is required, unless they happen to be made so as to be self-starting, then they are used in small sizes.

A synchronous motor, when running, will keep in step with the current, no matter how much the load may vary, provided it is not made greater than the capacity of the machine. If the load is made so great that the motor cannot carry it, the armature will be pulled out of step with the current and will immediately come to a stop. On this account, motors of the synchronous type are not well adapted to operate cranes or similar machines in which there is a liability of greatly overloading the machine occasionally.

The current developed by an alternating current generator will lag behind the impressed e.m.f. as has been fully explained in the foregoing. If this current is passed through a second machine, that acts as a motor, the latter will tend to generate a current that flows in opposition to that of the generator ; hence, in this current the lag will be in the opposite direction of that of the current that drives it. That is when the machine acts as a motor its whole action as a generator is reversed. Owing to this fact, if a synchronous motor is placed at one end of a circuit, and a generator at the other, the motor will act to neutralize the self-induction of the generator, and thus to bring the current in the circuit, and the impressed e.m.f. into phase with each other. Thus, a synchronous motor can be made to act in the same way as a condenser, to reduce the lag of the current.

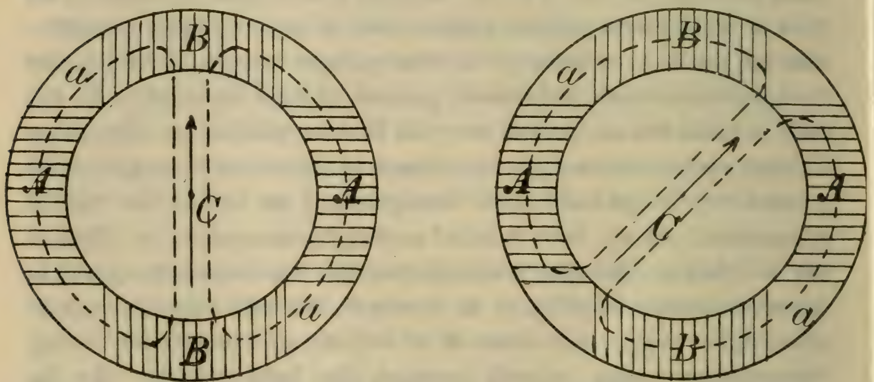
Power factor.— In an alternating current circuit, it is very important to reduce the lag of the current as far as possible because the actual amount of energy carried by the current depends upon the angle of lag, as was fully explained in connection with Figs. 372 to 374. In a continuous current circuit the power is always equal to the product of the volts by the amperes, but in an alternating current circuit this product is not a measure of the power. It is called the apparent power, or the volt-amperes. The actual power is equal to the amperes multiplied by the e.m.f. in phase with the current, or the active voltage, as it is called. The ratio between the true power and the volt-amperes is called the power factor. The power factor can be obtained by dividing the true power by the volt-amperes, and it may range from 100 per cent when the current and impressed e.m.f. are in phase down to five or ten per cent when the angle of lag is nearly 90 per cent. In actual working circuits the power factor ranges between about 95 and 75 per cent. Any kind of device that has a low reactance, as, for example, incandescent lamps, acts to keep the angle of lag of the current small, and thus

the power factor high. Devices having large reactance, such as transformers, and induction motors act to increase the angle of lag of the current, and thus to reduce the power factor. Devices that develop a negative reactance, that is, which cause the current to lead the impressed e.m.f., such as condensers and synchronous motors, can be used in circuits in which transformers and similar devices are operated so as to counteract these and thereby keep up the percentage of the power factor.

Induction and other types of motors. — In addition to the synchronous motors just explained, the only type of machine that requires notice here is the induction motor. This is by far the most extensively used of all alternating current motors, and from the manner in which it acts it has a greater range of adaptability than any other type. It may be well to mention here, however, that a plain motor, such as those used with continuous currents, can be made to operate with alternating currents providing the field cores are made laminated, instead of solid castings. If the field is solid the motor will not run if connected in an alternating current circuit because the large mass of iron constituting the field cannot be magnetized and demagnetized as fast as the current alternates. If we take hold of a freight car and try to shake it we will fail in the effort, simply because the bulk is too great to be set in motion rapidly. If, however, we take hold of the side of a light buggy and shake it we will be able to produce a very vigorous movement, simply because the bulk is light. In the same way, if we attempt to alternate the magnetic polarity of large masses of iron we fail because the bulk is too great, but if we divide the mass up into many thin sheets we will have no difficulty in causing its polarity to change rapidly. Alternating current motors of this kind which are called commutator motors, have been made, but they are not used or manufactured for commercial purposes at the present time, because they are far inferior to other types. They are open to two objections, one of which

is that they spark considerably and the other is that they will not give much more than one-third the power that the same machine will develop if supplied with a continuous current. The reason why they give such small power is that on account of the many turns of wire on the field the inductive action is very great, hence the reactance is very high, and as a result the current lags excessively so that the power factor is very low, therefore, although the current is strong, the actual energy carried by it is comparatively small. Several other types of alternating current motors have been devised, but they have never got beyond the experimental stage.

Principle of the induction motor.—Induction motors are made for single and polyphase currents. When in operation the



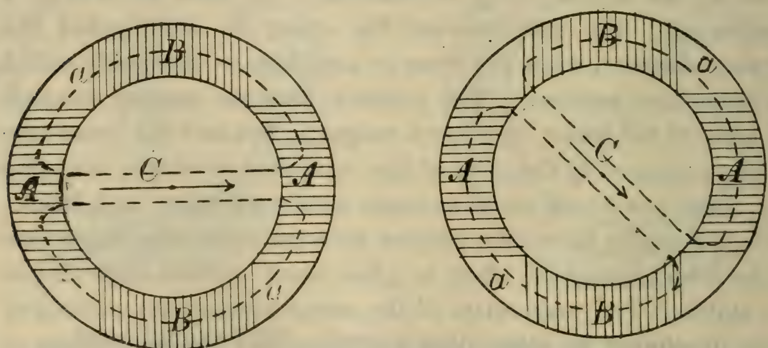
Figs. 392 and 393. Principle of induction motor.

principle of action is the same in all, but in the act of starting the single-phase machine is different from the others. Single-phase induction motors will not start of their own accord unless provided with special starting arrangements. The most common way of arranging a single-phase induction motor so as to be self-starting is to provide a set of starting coils that virtually convert it into a

two-phase machine in the act of starting. When the motor is under way the starting coils are cut out, although in some cases they are left in circuit all the time. The principle of the induction motor can be explained by the aid of the diagrams Figs. 392 to 395. These diagrams illustrate the action in a two-phase machine, which is the one most easily understood. The single-phase induction motor is the most difficult one to explain or to understand, so we will leave it for the last. In an induction motor, the stationary part, which is called the stator, and sometimes the field, is provided with coils that are connected with the operating circuits. The rotating part, which is called the rotor, and sometimes the armature, is provided with coils that are short circuited upon themselves and are not connected with the operating circuits. The principle of operation generally stated is that the currents in the stator develop an inductive action upon the coils of the rotor thus developing currents in these, the action being substantially the same as that in a transformer. On that account the stator is also called the primary member, while the rotor or armature is commonly called the secondary member. The primary currents passing through the coils of the stator, develop a magnetic flux and the secondary currents induced in the coils of the rotor also develop a magnetic flux, these two fluxes are at an angle with each other, and, hence, there is a strong attraction exerted between them, the magnetism of the rotor making an effort to place itself parallel with that of the stator. The magnetism of the stator rotates, on account of being developed by alternating currents, and the magnetism of the rotor in trying to place itself parallel with that of the stator also rotates, chasing the latter around the circle but never overtaking it.

In Fig. 392 let *A A* represent two coils connected in one of the circuits of a two-phase system, and let *B B* represent two other coils connected in the other circuit of this same system. Suppose

we consider the instant of time when the current flowing through the A A coils is at its maximum value, then at this very same instant the current in the B B coils will be zero. The current in the A A coils is then the only magnetizing current acting upon the ring at this instant. Suppose the direction of the current through A A is such as to develop a magnetic flux that will traverse the space in the center of the ring in the direction of arrow C. As the current in the A A coils begins to decrease, that flowing in the B B coils will begin to increase. Let the direction of the current in the B B coils be such as to send a magnetic flux through the center of the ring in the direction of arrow C in Fig. 394. This magnetization will act upon that developed by the current in the A A coils and will have a tendency to twist it around into the direction of arrow C in Fig. 393. When the current in the A A coils has reduced and the current in the B B coils has increased until they are both equal, then each one will act with equal force



Figs. 394 and 395. Illustrating operation of induction motor.

to establish a magnetization in its own direction, and the result will be that the actual direction of the magnetic flux will be as indicated by arrow C in Fig. 393. Thus we see that by the decrease in the strength of the current in the A A coils and the

increase in the strength of the current in the *B B* coils until they are both equal, the magnetic flux has been rotated from the position of arrow *C* in Fig. 392 to its position in Fig. 393. Now as the variation in the currents progresses, and that in *A A* becomes weaker, while that in *B B* becomes stronger, the direction of the magnetic flux will be still further rotated so that when the current in *B B* reaches the maximum value, and that in *A A* becomes zero, the direction of the flux will be that of arrow *C* in Fig. 394. As we advance beyond this instant of time, the current in *B B* will begin to reduce, while that in *A A* will begin to increase, but its direction will be the opposite of what it was in Fig. 392, so that when the currents in the two sets of coils become equal again, the direction of the magnetic flux will be that of arrow *C* in Fig. 395. When the current in the *A A* coils reaches the maximum and that in *B B* becomes zero, the flux will have rotated through one-half of a revolution and arrow *C* will be in the vertical position but pointing downward.

If we follow the action of the currents further we will find that as a result of the continuous increasing and decreasing and changing of direction, the magnetic flux indicated by arrow *C* will continuously rotate keeping time with the frequency of the currents. Now if we suppose that an armature upon which a number of coils are wound in a diametrical position, is placed within the field ring, and is held stationary, we will see at once that the rotating magnetic flux will cut through its coils and develop e.m.fs. in them. The currents developed in these coils on the stationary armature will be alternating, hence, they will develop a magnetic flux in the armature that will rotate, and keep time with the rotating flux developed by the field coils. Both these fluxes act inductively upon the field and armature coils, their combined effect being equal to that of a single flux located 90 degrees in advance of the e.m.f. induced in the armature coils, hence, somewhat more than 90 degrees ahead of

the armature current. If we hold the armature by means of a brake, and free this slightly, so that the armature may revolve slowly, it will at once follow around after the rotating field, but as its magnetization is developed by currents that are induced by the action of the field magnetism, it will matter little how fast the armature may revolve, its magnetization will never be able to overtake that of the field.

As can be judged from the foregoing explanation, an induction motor is not a synchronous machine, and its armature can never attain a velocity equal to that of the rotating field. If the resistance of the armature coils is made very low, it may reach a velocity very near to that of the rotating flux. The difference between the velocity of the rotating flux and that of the rotating armature is called the slip of the motor.

If the motor is designed for constant speed, the resistance of the armature coils is made very low, and then when the machine is running free, the speed of the armature may run up to 99 or 99½ per cent of the speed of the rotating field, and when the maximum load is put on it may not drop lower than 94 or 95 per cent. If a motor is designed in this way the pull of the armature when it starts up will be small and will gradually increase until the speed is about nine-tenths of the maximum when it will again begin to decrease.

If it is desired to make a motor that will give a strong pull when it starts up, its armature coils must have more resistance, and then it will pull harder on the start, but as fast as the speed builds up the pull will reduce. From this it will be seen that induction motors that are made so as to run at nearly a constant speed, say to vary five or six per cent between full load and running free, will not give a strong pull in the act of starting, hence they will have to be started without a load. If a motor is to be made to start under a full load it must be proportioned so that it will not run at a constant speed, but will gradually reduce its velocity as the load is increased.

Induction motors, if very small, are started by connecting them directly with the operating circuits, but if they are of any capacity they must be provided with some kind of starting resistance so as to keep the starting current down within safe limits. One way of starting is to introduce resistance into the primary circuits, but this results in reducing the strength of the field, and thus the pull of the armature. Another way is to introduce resistance into the armature coil circuit. This is the best method, because it enables the motor to start up with a strong pull.

Three-phase induction motors act in precisely the same way as the two-phase, the only difference being that the rotation of the field flux is produced by the increase and decrease in the strength of three currents flowing through three sets of coils equally spaced around the circle instead of by the increase and decrease in two currents flowing in two sets of coils equally spaced around the circle.

In the single-phase induction motor, the magnetic flux developed by the single alternating current traversing a single set of coils on the field combines with the magnetic flux developed by the armature current, to develop a rotating field and this acting upon the armature coils produces rotation in precisely the same way as in the two-phase machine. This is the action that takes place after the armature is set in motion, but if the load is increased and the armature speed is reduced the rotating field begins to become irregular, and by the time the armature velocity is reduced to about one-half, the rotating flux becomes so irregular in its movement, that the armature pull begins to reduce very rapidly, and the machine comes to a standstill. Owing to this fact single-phase induction motors cannot be used in cases where it is desired to start with a strong pull, or where a wide range of speed variation is desired.

To make a single-phase induction motor self-starting, it is wound

with two sets of coils, like the diagrams Figs. 392 to 395, and the current from the single-phase circuit is passed through these two sets of coils in parallel branches, and in one of the branches the reactance is greatly increased, so as to make the current in this branch lag much more than in the other. In this way a phase displacement is obtained between the two currents, and this produces a corresponding displacement in the magnetic fluxes developed by the two sets of coils, so that their combined action develops a rotating field. This field does not rotate at a uniform rate, like the field of a two-phase motor, but it is uniform enough for the purpose of setting the machine in motion. To increase the reactance in the auxiliary starting coils, all that is necessary is to wind them with many turns of fine wire, and this is an arrangement very commonly employed, but, in some cases, separate coils are placed in the auxiliary circuit to obtain the required reactance.

There are other ways in which single-phase induction motors are made self-starting, but they are not very extensively used.

While induction motors are very satisfactory machines, being adapted to every kind of work, even to the operation of railway cars, they have the objection of being highly inductive devices that act to greatly increase the lag of the current, and thereby to reduce the power factor. On this account they are often used in connection with synchronous motors so that the latter may counteract their inductive effect, and thus keep the power factor high.

The small motor shown in Fig. 391, is an induction motor. Induction motors are made in many different designs, and as large as 300 to 400 H. P., but as a rule they are confined to much smaller capacities; synchronous motors being used for the larger sizes.

Rotary transformers and rotary converters. — A rotary transformer is a machine by means of which a continuous current may be obtained from an alternating current. A rotary con-

verter is a machine for accomplishing the same result. The essential difference between the two is that the first is driven by an alternating current and generates a continuous current, while the second changes an alternating into a continuous current. As a result of this difference the rotary transformer can be used to obtain a continuous current of any desired voltage from an alternating current of any given voltage; but in the rotary converter, as the action is to convert the alternating into a continuous current, the voltage relation is fixed so that for a given alternating current voltage we will get a corresponding continuous current voltage. Both these machines can be used in the reverse order, that is to transform or convert a continuous into an alternating current.

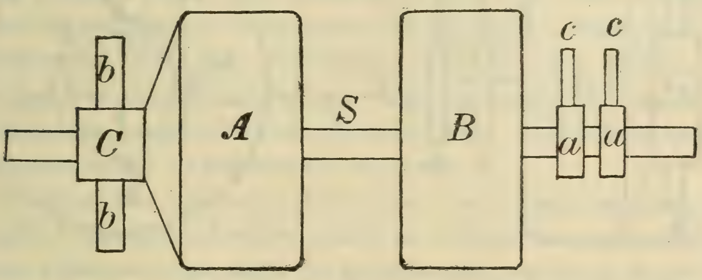
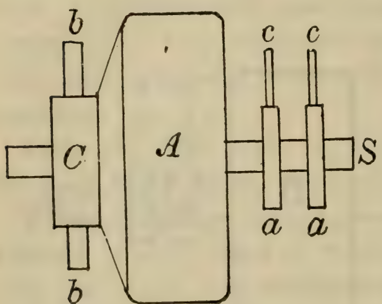
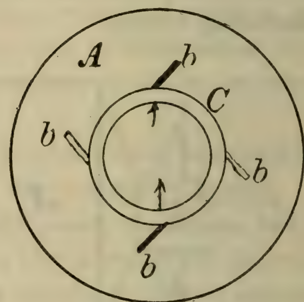


Fig. 396. Principle of the rotary transformer.

Principle of the rotary transformer.—The principle of the rotary transformer is illustrated in Fig. 396. In this diagram A represents a continuous current armature, and B is an alternating current armature. If both these are provided with suitable magnetic fields then if continuous current is passed through A it will become a motor and will drive B and generate therein a single alternating current or a number of them according to the way in which the armature is wound. Thus B may become a single or a polyphase generator. It can further be seen that the

voltage of the currents generated by *B* is in no way connected with the voltage of the current that drives *A*, and depends wholly upon the way in which *B* is wound. If *B* is connected with an alternating current circuit, then it will run as a synchronous motor and drive *A* and the latter will generate a continuous current. This machine if driven by a continuous current will be self-starting, but if driven by an alternating current it will have to be started. If driven by an alternating current its speed will be controlled by the frequency of the current, but if driven by a continuous current its speed will vary with the magnitude of the load placed upon it.





Figs. 397 and 398. Principle of the rotary converter.

Figs. 397 and 398 illustrate the principle of operation and the construction of a rotary converter. The armature *A* is of the continuous current type, having a commutator *C*. If it is a two-pole machine, then if wires are connected with diametrically opposite segments of the commutator as is indicated in Fig. 398 by the arrows, and these are connected with the collector rings *a a*, brushes *c c* placed on these rings, will take of a true alternating current if the armature is placed in a suitable field and is driven. While alternating current can be taken from the brushes *c c*, a continuous current can also be taken from the brushes *b b*,

which bear upon the commutator *C*. Thus, this machine, if driven, becomes a combination generator which will deliver a continuous and an alternating current at the same time. Machines of this type are constructed and are called double current generators.

If the brushes *c c* are connected with a single-phase circuit, and the armature is placed in a suitable field, it will rotate and from the *b b* brushes of the commutator a continuous current can be drawn. If the brushes *b b* are connected with a continuous current circuit, an alternating current will be delivered through the brushes *c c*.

If four wires are connected with four commutator segments one quarter of the circumference apart, and these are connected with four collector rings, then from these rings two alternating currents 90 degrees out of phase can be obtained. Thus, with four connections with the commutator segments the machine can convert two-phase currents into one continuous current, or one continuous current into two-phase currents, that is into two alternating currents 90 degrees out of phase.

If wires are connected with three commutator segments one-third of the circumference apart, and these are connected with three collector rings, then the machine will become a three-phase converter, and if connected with a three-phase system will deliver one continuous current or if connected with a continuous current circuit will deliver the three currents of a three-phase system.

The rotary converter, as will be seen from the foregoing, actually changes a continuous current into one or more alternating currents, or one or more alternating currents into one continuous current, and in every case there is a direct electrical connection between the continuous and the alternating current circuits. As this type of machine simply converts the current of one type into current of the other type it is quite evident that there must be a fixed relation between the strength of the alternat-

ing and continuous currents and also between the voltages. An alternating current if of the sine type, will have an effective value of 70.7 per cent of its maximum value, for the amperes as well as the volts. So that if we have a continuous current of 70.7 amperes and 70.7 volts, we must have an alternating current of 100 amperes maximum value and 100 volts maximum value to be equal to it, and if the energy is also to be equal, the current in the alternating current circuit must be in phase with it e.m.f., that is the power factor must be 100.

In a rotary converter the voltage of the continuous current is equal to the maximum voltage of the alternating current and the strength of the continuous current is equal to one-half the maximum strength of the alternating current. Thus if the maximum voltage of the alternating current is 1,000 volts, the voltage of the continuous current will be 1,000, and if the maximum strength of the alternating current is 100 amperes the strength of the continuous current will be 50 amperes. This arises from the fact that the rotary converter does not develop energy, as it drives itself, hence, the energy in the continuous current cannot be more than that in the alternating, in fact it will be a trifle less owing to the energy absorbed in driving the machine. Now if the alternating e.m.f. and current have the maximum values of 1,000 volts and 100 amperes, their effective values will be 707 volts and 70.7 amperes, and the product of these two will be the energy in watts. Thus $707 \times 70.7 = 50{,}000$ watts. Now if the voltage of the continuous current is 1,000, its strength must be 50 amperes, less the amount absorbed in overcoming the friction of the machine.

Fig. 399 shows a rotary converter of large size.

Alternating current distributions. — The principal advantage of alternating over continuous currents is that they can be used for transmitting energy to much greater distances, owing to the fact that a high voltage can be used to transmit the main current over the wire, and at the receiving end this current can be passed

through transformers, from which secondary currents of low voltage may be obtained. In a few instances, low voltage alternating

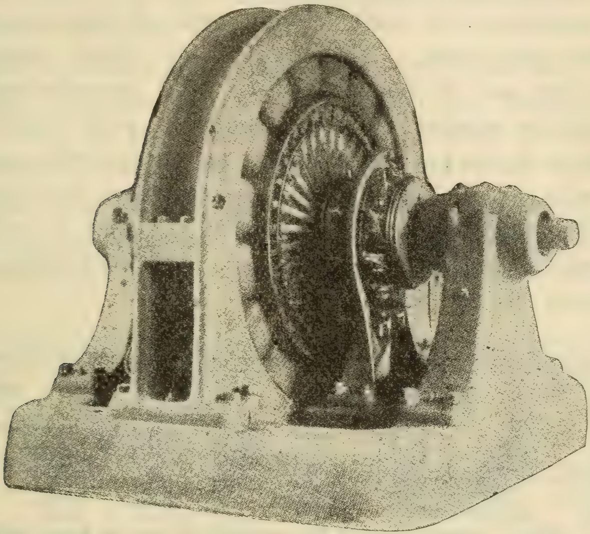


Fig. 399. Rotary converter.

currents are used for distributing current over small areas. The general arrangement of circuits and apparatus for a three-phase system of this kind is illustrated in the diagram Fig. 400.

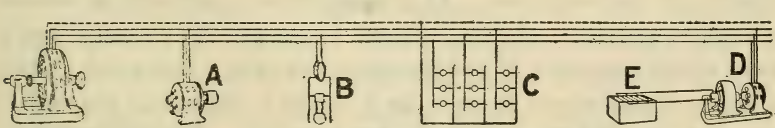


Fig. 400. General arrangement of three-phase system.

The generator is shown at the extreme left. At *A* an induction motor is connected with the circuit. At *B* an "arc" light is connected in the secondary circuit of a small transformer. At *C*

a number of incandescent lamps are connected. At *D* the circuit is used to drive a rotary transformer, which develops a continuous current to charge storage batteries at *E*. The three solid line wires constitute the main circuit and all the apparatus is connected with them. The broken line above these is the neutral wire and is connected with the incandescent lamps only. If the number of these lamps in each circuit is the same, as is shown on the diagram, no current will pass to the neutral wire, but if in one of the circuits there are more lamps than in the other, the excess of current will pass to or from the neutral wire. Systems of this type are operated at voltages ranging between 200 and 600.

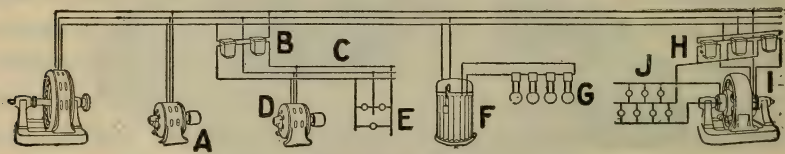


Fig. 401. Arrangement for distances up to four miles.

The diagram, Fig. 401, shows the way in which the circuits are arranged when the distance of transmission is from one to three or four miles. For such cases, the voltage generally used is 2300. The generator at the left develops currents that pass directly to the main line. At *A* an induction motor is connected directly to the main line. At *B* transformers are used to develop secondary currents of low voltage to supply the circuit wires *C* from which the motor *D* and incandescent lamps *E* are fed. At *F* a series transformer is used to develop a secondary current of constant strength to operate the arc lamps *G*. The difference between a series transformer and the ordinary type is that the former is provided with a mechanical regulator, actuated by the current which maintains the secondary current of constant strength and varies the voltage in accordance with the number of

lamps in service. At H another set of transformers are used to develop low voltage secondary currents, which pass through a rotary converter I, and are converted into a continuous current to feed the incandescent lamps at J.

Fig. 402 illustrates the arrangement of circuits and apparatus for long distance transmissions, which may range all the way from five or six miles up to one hundred or more, the greatest distance covered up to date being 145 miles. To transmit current to great distances with a small loss in the transmission lines, it is necessary to use very high voltages, ranging from 10,000 to 60,000, and as it is not advisable to construct

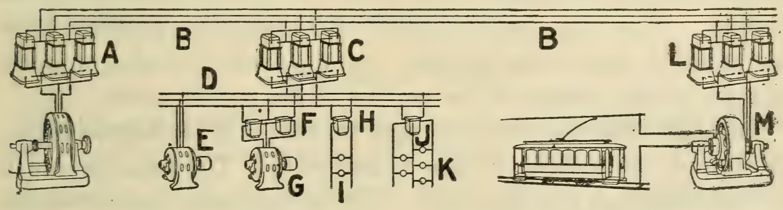


Fig. 402. Arrangement for long distances.

generators to develop such high pressures, raising transformers are employed to develop the line current. These transformers are shown in Fig. 402 at A. The generator develops currents at 1,000 volts, and this passing through the primary coils of the transformers at A induces secondary currents which may have any voltage desired, say, 20,000. These secondary currents pass to the transmission lines B B, which may extend a distance of ten, twenty or more miles and may deliver all their energy at the end of the line or drop part of it at intermediate points. The transformers at C and also those at L develop secondary currents of any lower voltage that may be required; thus, those at C develop secondary currents for the circuits D, which may be of, say, 1,000 volts. The motor E is shown connected directly with D, but

motor G and lamps I, K require a still lower voltage, hence the currents in D are passed through a second set of transformers at F, H and J. The three transformers at L develop secondary currents of sufficiently low voltage to be passed through the rotary converter M, and thus provide a continuous current for the trolley road as shown.

STARTING.

When the armature is turning, see that the oil rings in the bearings are in motion. When the machine is up to speed and all switches are open, lower the brushes on the commutator and collector, making sure that each bears evenly and squarely on the surface. Turn the rheostat until all resistance is in, then close the switch in the exciter circuit. Set the exciter brushes properly and adjust the voltage of the exciter to the proper point.

The alternator rheostat may then be turned gradually over until the proper alternating voltage is indicated. The main circuit of the machine may now be closed. The commutator brushes should be adjusted at a non-sparking position. If there is any load the voltage should increase slightly. If it decreases, it shows that the series coils and the separately excited coils are opposing each other, unless this decrease is caused by a drop in speed. If it is found that the coils are opposing each other, unclamp the brush-holder yoke of the alternator and move its commutator brushes backward or forward one and one-half segments in a three-phase machine, and one segment in a two-phase machine. A position giving maximum voltage will be found from which any motion, forward or backward, diminishes the voltage. Having once determined the correct setting of the brushes, they may generally remain unchanged, unless the generator is subject to great variation of load when in some machines, slight movements may be found desirable.

PARALLEL RUNNING OF ALTERNATORS.

TYPES SUITABLE FOR PARALLEL OPERATION.

If the speeds are exactly adjusted, any two alternators of the same frequency will operate together in parallel. The maximum angular displacement that may take place between two machines in parallel without causing objectionable phase difference decreases with increased number of poles. For this reason high frequencies are, generally speaking, less favorable to parallel operation than lower frequencies. Machines of the highest frequencies ordinarily used can, however, be successfully run in parallel if the mechanical arrangements are suitable.

DIVISION OF LOAD.

Machines to operate in parallel must run at such speeds as will give exact equality of frequency. If the prime mover running one machine tends to produce a lower frequency than that running the other, the machines cannot carry equal loads.

When two alternators operate in parallel, each must carry an amount of load proportionate to the power received from its prime mover. If one engine or water-wheel governs in such a manner as to give more power than the other, this machine must carry more load, no matter what the field excitation may be. If under such conditions the field excitations are correct, both machines will deliver current to the line in approximately the proportions in which they receive power from their prime movers. If the field adjustments are incorrect, there will be idle currents between the machines in addition to the currents which go to the line.

COMPOUND ALTERNATORS.

When compound alternators are operated in parallel, equalizer connections should be used so that the rectified alternating

current can properly distribute itself into the fields of all the machines. Without equalizers, an unstable condition may exist which will render parallel operation unsatisfactory. This applies particularly in the case of machines driven from the same source of power. The greater the amount of compounding, the greater will be the tendency to instability.

BELTED MACHINES.

If two machines are belted to separate prime movers, their parallel operation is dependent upon the governing of the prime movers. If they are belted to the same source of power, their parallel operation depends upon the proportions of pulleys and belts, and upon the tension and friction of the latter. Under such conditions the pulleys and belts must be adjusted with great nicety, so that both machines will tend, with proper belt tension, to run at exactly the same frequency. Even where pulleys are of exactly the correct dimensions, a slight difference in the thickness of belts may cause considerable cross currents or unequal division of load.

DIRECT COUPLED MACHINES.

With such machines, engines must not only be adjusted to run at synchronous speed, but must also be provided with fly-wheels large enough to prevent appreciable variations of frequency within each revolution. Inequalities of speed, due to insufficient fly-wheel effect, will cause periodic cross currents between dynamos, or will entirely prevent their operation in parallel. The greater the number of poles in a direct coupled machine, the less the angular speed variation necessary to cause trouble.

High speeds are much more desirable with direct coupled alternators than low speeds, and low frequencies present less diffi-

culties than high. The desirabilty of high speeds with direct coupled alternators cannot be too strongly stated. While an increase of fly-wheel effect will equalize the angular irregularities of an engine's motion, it cannot bring about such good results as would be brought about by a similar reduction of angular error effected through an increase of speed. While the large fly-wheel steadies the motion, it may tend to prevent correction of the angular error through the effect of the cross currents. Cross currents, which flow in machines having light fly-wheels may have an effective tendency to hold them together; while machines with very heavy fly-wheels may tend to act independently of each other as far as angular variations are concerned.

These matters should be carefully considered in installing direct connected alternators. Where engines operate at the same speed and have the same number of cranks, this trouble can sometimes be overcome by synchronizing the engines themselves so that the impulse in both come together. When the fly-wheel effect is insufficient, the frequency will fluctuate and this fluctuation may cause serious trouble if synchronous motors or rotary converters are connected to the circuit. When the cranks of two engines coupled to alternators are synchronized, any fluctuation of frequency which is due to lack of fly-wheel effect will still exist, although it may not affect parallel running.

Where alternators have to be operated in parallel by engines to which they are directly coupled, it is generally desirable to use engines having as many cranks as possible, so that the crank efforts will be well distributed throughout the revolution, and will not tend to produce an irregularity of motion.

STARTING.

When a machine driven by a separate engine is thrown in parallel with others which are carrying load, the throttle should be partly closed so that it can just run at synchronous speed with-

out carrying load. After it is in step with the other machines, load can gradually be taken on by giving it more steam. If this is carefully done the voltage on the circuit is not disturbed by the addition of the new machine.

When a belted machine is to be thrown into parallel with others driven by the same shaft, its belt tension should first be reduced, which will tend to admit enough slip to bring it into step with the loaded machines. After it is thrown in it will gradually take load as the belt is tightened.

SHUTTING DOWN.

In shutting down machines operating singly, both the generator and exciter field resistance should be cut in by turning the rheostat before the line switch is opened.

When two or more generators are running in parallel on the bus-bars, one may be shut down at any time. The equalizer switch should be opened *first*, then the load reduced by throttling the engine or by slacking the belt. As soon as the load is practically off, open the main switch.

CARE OF MACHINES.

With high voltage machines it is absolutely essential that they be kept scrupulously clean. Small particles of copper or carbon dust, may be sufficient to start a disastrous arc.

The commutator collector should receive careful attention and be wiped thoroughly every day.

From time to time the machine should be thoroughly overhauled and given a coating of air-drying japan after cleaning. Machines of the rotary field type are so constructed that it is a comparatively easy matter to get at every part of the armature coils. In a large station it is recommended that an air compressor be installed so that a hose can be led to the machine and the dust thoroughly blown out.

It is advisable to have rubber mats in front of high tension switchboards and on the floor at the commutator-collector end of the generator. If it is necessary to adjust the brushes while the machine is in operation, the attendant should stand on the mat and it is also recommended that he wear rubber gloves.

Both commutator and collector rings require a very slight amount of vaseline. In applying it a dry stick with a little chamois leather tied to one end may be used, so that there will be no danger of coming in contact with the brushes.

With the brushes properly set and all screws firmly tightened into place, the generators should require very little attention while running. It is well to note from time to time whether the oil rings are working properly.

Electrical Formulas.

$$\text{Watts} = \text{Amperes} \times \text{Volts}. \quad \text{Watts} = \frac{\text{Volts} \times \text{Volts}}{\text{Ohms}}$$

$$\text{Watts} = \text{Amperes} \times \text{Amperes} \times \text{Ohms}.$$

$$\text{Volts} = \text{Amperes} \times \text{Ohms}. \quad \text{Amperes} = \frac{\text{Volts}}{\text{Ohms}}$$

$$\text{Ohms} = \frac{\text{Volts}}{\text{Amperes}} \quad \text{Heat units per Sec.} = \text{Amp.} \times \text{Amp.} \times \text{Ohms} \times \text{Secs.} \times 0.24$$

$$\text{Electrical Horsepower} = \frac{\text{Watts}}{746}$$

$$\text{H. P. lost in conductor} = 16.6538 \times \left(\frac{\text{H. P.}}{\text{Initial Volts}}\right) \times \text{length in miles.}$$

$$\text{Area of cond. in circular mils} = \frac{2150 \times \text{Watts per lamp}}{\text{Volts} \times \text{Volts} \times \%\text{ of drop}} \times \text{no. of lamps} \times \text{dist. to center of distribution in miles, or}$$

$$\frac{2150 \times \text{dist. to center of distribution} \times \text{Amperes}}{\%\text{ of Drop} \times \text{Volts}}$$

$$\text{Weight of copper} = \frac{\text{Distance in miles} \times \text{Distance in miles}}{\text{Volts} \times \text{volts} \div 100} \times$$

$$\text{H. P. delivered to motor} \times \frac{100 - \%\text{ of line loss}}{\%\text{ of line loss}} \times 266.5$$

Energy Required to Produce 1 Candle Power.

	Watts.		Watts.		Watts.
Tallow	124	Mineral oils	80	Cannel gas	48
Wax	94	Vegetable oils	57	Incandescent lamp	15
Spermaceti	86	Coal gas	68	Arc lamp	3

CHAPTER XXX.

A CHAPTER OF TABLES.

TABLE NO. 1.—HYPERBOLIC LOGARITHMS.

Ratio	Log.	Ratio	Log.	Ratio	Log.	Ratio	Log.	Ratio	Log.	Rotio	Log.	Ratio	Log.
1.00	.0000	2.45	.8961	3.90	1.3610	5.35	1.6771	6.80	1.9169	8.25	2.1102	9.70	2.2721
1.05	.0488	2.50	.9163	3.95	1.3737	5.40	1.6864	6.85	1.9242	8.30	2.1163	9.75	2.2773
1.10	.0953	2.55	.9361	4.00	1.3863	5.45	1.6956	6.90	1.9315	8.35	2.1223	9.80	2.2824
1.15	.1398	2.60	.9555	4.05	1.3987	5.50	1.7047	6.95	1.9387	8.40	2.1282	9.85	2.2875
1.20	.1823	2.65	.9746	4.10	1.4110	5.55	1.7138	7.00	1.9459	8.45	2.1342	9.90	2.2925
1.25	.2231	2.70	.9933	4.15	1.4231	5.60	1.7228	7.05	1.9530	8.50	2.1401	9.95	2.2976
1.30	.2624	2.75	1.0116	4.20	1.4351	5.65	1.7317	7.10	1.9601	8.55	2.1459	10.00	2.3026
1.35	.3001	2.80	1.0296	4.25	1.4469	5.70	1.7405	7.15	1.9671	8.60	2.1518		
1.40	.3365	2.85	1.0473	4.30	1.4586	5.75	1.7402	7.20	1.9741	8.65	2.1576		
1.45	.3716	2.90	1.0647	4.35	1.4702	5.80	1.7579	7.25	1.9810	8.70	2.1633		
1.50	.4055	2.95	1.0818	4.40	1.4816	5.85	1.7664	7.30	1.9879	8.75	2.1691		
1.55	.4383	3.00	1.0986	4.45	1.4929	5.90	1.7750	7.35	1.9947	8.80	2.1748		
1.60	.4700	3.05	1.1151	4.50	1.5041	5.95	1.7834	7.40	2.0015	8.85	2.1804		
1.65	.5008	3.10	1.1314	4.55	1.5151	6.00	1.7918	7.45	2.0082	8.90	2.1861		
1.70	.5306	3.15	1.1474	4.60	1.5261	6.05	1.8001	7.50	2.0149	8.95	2.1917		
1.75	.5596	3.20	1.1632	4.65	1.5369	6.10	1.8083	7.55	2.0215	9.00	2.1972		
1.80	.5878	3.25	1.1787	4.70	1.5476	6.15	1.8165	7.60	2.0281	9.05	2.2028		
1.85	.6152	3.30	1.1939	4.75	1.5581	6.20	1.8245	7.65	2.0347	9.10	2.2083		
1.90	.6419	3.35	1.2090	4.80	1.5686	6.25	1.8326	7.70	2.0412	9.15	2.2138		
1.95	.6678	3.40	1.2238	4.85	1.5790	6.30	1.8405	7.75	2.0477	9.20	2.2192		
2.00	.6931	3.45	1.2384	4.90	1.5892	6.35	1.8485	7.80	2.0541	9.25	2.2246		
2.05	.7178	3.50	1.2528	4.95	1.5994	6.40	1.8563	7.85	2.0605	9.30	2.2300		
2.10	.7419	3.55	1.2669	5.00	1.6094	6.45	1.8641	7.90	2.0669	9.35	2.2354		
2.15	.7655	3.60	1.2809	5.05	1.6194	6.50	1.8718	7.95	2.0732	9.40	2.2407		
2.20	.7885	3.65	1.2947	5.10	1.6292	6.55	1.8795	8.00	2.0794	9.45	2.2460		
2.25	.8109	3.70	1.3083	5.15	1.6390	6.60	1.8871	8.05	2.0857	9.50	2.2513		
2.30	.8329	3.75	1.3218	5.20	1.6487	6.65	1.8946	8.10	2.0919	9.55	2.2565		
2.35	.8544	3.80	1.3350	5.25	1.6582	6.70	1.9021	8.15	2.0980	9.60	2.2618		
2.40	.8755	3.85	1.3181	5.30	1.6677	6.75	1.9095	8.20	2.1041	9.65	2.2670		

To find the hyperbolic logarithm of a ratio which is ten times any ratio given in the table, find the ratio in the table which is one-tenth of the given ratio and add 2.3026 to the corresponding logarithm; the sum will be the required logarithm. Example: What is the hyperbolic logarithm of 15.5? 15.5 is ten times 1.55. The logarithm of 1.55 is .4383, and .4383 + 2.3026 = 2.7409 which is the hyperbolic logarithm of 15.5.

TABLE NO. 2.

DUPLEX STEAM PUMPS.

For Water Pressure Not Exceeding 150 lbs. Speed from 50 to 100 feet per Minute.

Diameter of Steam Cylinders.	Diameter of Water Plungers.	Length of Stroke.	Displacement in Gallons per stroke of one Plunger.	Proper Strokes per minute of one Plunger, varying with kind of work and Pressure.	Gallons delivered per minute by both Plungers, at stated number of Strokes.
3	2	3	.04	100 to 250	8 to 20
4½	2¾	4	.10	100 to 200	20 to 40
5¼	3½	5	.20	100 to 200	40 to 80
6	4	6	.33	100 to 150	70 to 100
7½	4½	6	.42	100 to 150	85 to 125
7½	5	6	.51	100 to 150	100 to 150
7½	4½	10	.69	75 to 125	100 to 170
9	5¼	10	.93	75 to 125	135 to 230
10	6	10	1.22	75 to 125	180 to 300
10	7	10	1.66	75 to 125	245 to 410
12	7	10	1.66	75 to 125	245 to 410
14	7	10	1.66	75 to 125	245 to 410
12	8½	10	2.45	75 to 125	365 to 610
14	8½	10	2.45	75 to 125	365 to 610
16	8½	10	2.45	75 to 125	365 to 610
18½	8½	10	2.45	75 to 125	365 to 610
20	8½	10	2.45	75 to 125	365 to 610
12	10¼	10	3.57	75 to 125	530 to 890
14	10¼	10	3.57	75 to 125	530 to 890
16	10¼	10	3.57	75 to 125	530 to 890
18½	10¼	10	3.57	75 to 125	530 to 890
20	10¼	10	3.57	75 to 125	530 to 890
14	12	10	4.89	75 to 125	730 to 1220
16	12	10	4.89	75 to 125	730 to 1220
18½	12	10	4.89	75 to 125	730 to 1220
20	12	10	4.89	75 to 125	730 to 1220
18½	14	10	6.66	75 to 125	990 to 1660
20	14	10	6.66	75 to 125	990 to 1660
17	10	15	5.10	50 to 100	510 to 1020
20	12	15	7.34	50 to 100	730 to 1460
20	15	15	11.47	50 to 100	1145 to 2290
25	15	15	11.47	50 to 100	1145 to 2290

TABLE NO. 3

Tank or Light Service Pumps.

These pumps are principally used at railroad water stations, gas and oil works, bleacheries, tanneries, refineries, plantations, distilleries, etc. A variety of valves are used adapted for pumping hot, cold, thick, thin, alkaline or other liquids.

For quarries and clay pits, also for coffer dams, tunnels, foundation pits, ore beds, sewerage and irrigating purposes, these pumps are especially adapted, having large water passages and valve openings.

SIZES AND CAPACITIES.

Steam Cylinder. Inches.	Water Cylinder. Inches.	Stroke. Inches.	Gallons per stroke.	Capacity per minute at ordinary speed.	Steam pipe. Ins.	Exhaust pipe. Inches.	Suction pipe. Inches.	Delivery pipe. Inches.	Floor Space Required. Inches.
3¼	3¼	4	.15	125 Strokes, 18 gals.	½	¾	1½	1¼	28 x10
4	4	5	.27	125 " 33 "	½	¾	2	1½	34 x11
5	4	7	.39	125 " 49 "	¾	1	2½	2	44 x12
5½	5½	7	.72	125 " 90 "	¾	1	3	2½	44 x13½
6	5½	7	.72	125 " 90 "	¾	1	3	2½	44 x13½
6	6	12	1.47	100 " 147 "	¾	1	4	4	66¾x19
6	7	12	2.00	100 " 200 "	¾	1	5	5	66¾x19
7½	7	10	1.66	100 " 166 "	1	1¼	5	5	56½x19
7½	7½	10	1.91	100 " 191 "	1	1¼	5	5	56½x19
8	6	12	1.47	100 " 147 "	1	1¼	4	4	66¾x19
8	7	12	2.00	100 " 200 "	1	1¼	5	5	66¾x19
8	8	12	2.61	100 " 261 "	1	1¼	5	5	66¾x20
8	9	12	3.30	100 " 330 "	1	1¼	6	6	66¾x21½
8	10	12	4.08	100 " 408 "	1	1¼	6	6	66¾x21½
10	10	12	4.08	100 " 408 "	1¼	1½	6	6	66¾x21½
10	10	16	5.44	75 " 408 "	1¼	1½	6	6	78½x21½
10	12	12	5.87	100 " 587 "	1¼	1½	8	6	66¾x23¼
10	12	16	7.83	75 " 587 "	1¼	1½	8	6	78½x23¾
12	10	12	4.08	100 " 408 "	2	2½	6	6	66¾x21½
12	10	16	5.44	75 " 408 "	2	2½	6	6	78½x21½
12	12	12	5.87	100 " 587 "	2	2½	8	6	66¾x23¾
12	12	16	7.83	75 " 587 "	2	2½	8	6	78½x23¾
14	12	12	5.87	100 " 587 "	2	2½	8	6	66¾x23¾
14	12	16	7.83	75 " 587 "	2	2½	8	6	78½x23¾
14	14	16	10.66	75 " 800 "	2	2½	10	8	78½x27
14	14	24	16.00	50 " 800 "	2½	3	10	8	108 x27
14	16	16	14.92	75 " 1020 "	2½	3	12	10	80 x35½
14	16	24	20.88	50 " 1044 "	2½	3	12	10	108 x35½
16	14	16	10.66	75 " 800 "	2½	3	10	8	78½x27
16	14	24	16.00	50 " 800 "	2½	3	10	8	108 x27
16	16	16	14.92	75 " 1020 "	2½	3	12	10	80 x35½
16	16	24	20.88	50 " 1044 "	2½	3	12	10	108 x35½
16	18	24	26.44	50 " 1322 "	2½	3	12	10	108 x38
16	20	24	32.64	50 " 1632 "	2½	3	14	12	108 x40
18	16	24	20.88	50 " 1044 "	3½	4	12	10	110 x35½
18	18	24	26.44	50 " 1322 "	3½	4	12	10	110 x38
18	20	24	32.64	50 " 1632 "	3½	4	14	12	110 x40
18	22	24	39.50	50 " 1975 "	3½	4	14	14	110 x42
20	18	24	26.44	50 " 1322 "	3½	4	12	10	118 x38
20	20	24	32.64	50 " 1622 "	3½	4	14	12	118 x40
20	22	24	39.50	50 " 1975 "	3½	4	14	14	118 x42
20	24	24	47.00	50 " 2350 "	3½	4	16	16	118 x44

TABLE NO. 4.

DIAMETERS, AREAS AND CIRCUMFERENCES OF CIRCLES.

Diam. Inches.	Circumf. Inches.	Area. Sq. In.	Diam. Inches.	Circumf. Inches.	Area. Sq. In.	Diam. Inches.	Circumf. Inches.	Area. Sq. In.
1	3.14159	0.78540	4	12.5664	12.566	8	25.1327	50.265
1/16	3.33794	0.88664	1/16	12.7627	12.962	1/8	25.5224	51.849
1/8	3.53429	0.99402	1/8	12.9591	13.364	1/4	25.9181	53.456
3/16	3.73064	1.1075	3/16	13.1554	13.772	3/8	26.3108	55.088
1/4	3.92699	1.2272	1/4	13.3518	14.186	1/2	26.7035	56.745
5/16	4.12334	1.3530	5/16	13.5481	14.607	5/8	27.0962	58.426
3/8	4.31969	1.4849	3/8	13.7445	15.033	3/4	27.4889	60.132
7/16	4.51604	1.6230	7/16	13.9408	15.466	7/8	27.8816	61.862
1/2	4.71239	1.7671	1/2	14.1372	15.904	9	28.2743	63.617
9/16	4.90874	1.9175	9/16	14.3335	16.349	1/8	28.6670	65.397
5/8	5.10509	2.0739	5/8	14.5299	16.800	1/4	29.0597	67.201
11/16	5.30144	2.2365	11/16	14.7262	17.257	3/8	29.4524	69.029
3/4	5.49779	2.4053	3/4	14.9226	17.721	1/2	29.8451	70.882
13/16	5.69414	2.5802	13/16	15.1189	18.190	5/8	30.2378	72.760
7/8	5.89049	2.7612	7/8	15.3153	18.665	3/4	30.6305	74.662
15/16	6.08684	2.9483	15/16	15.5116	19.147	7/8	31.0232	76.589
2	6.28319	3.1416	5	15.7080	19.635	10	31.4159	78.540
1/16	6.47953	3.3410	1/16	15.9043	20.129	1/4	32.2013	82.516
1/8	6.67588	3.5466	1/8	16.1007	20.629	1/2	32.9867	86.590
3/16	6.87223	3.7583	3/16	16.2970	21.135	3/4	33.7721	90.763
1/4	7.06858	3.9761	1/4	16.4934	21.648	11	34.5575	95.033
5/16	7.26493	4.2000	5/16	16.6897	22.166	1/4	35.3429	99.402
3/8	7.46128	4.4301	3/8	16.8861	22.691	1/2	36.1283	103.87
7/16	7.65763	4.6664	7/16	17.0824	23.221	3/4	36.9137	108.43
1/2	7.85398	4.9087	1/2	17.2788	23.758	12	37.6991	113.10
9/16	8.05033	5.1572	9/16	17.4751	24.301	1/4	38.4845	117.86
5/8	8.24668	5.4119	5/8	17.6715	24.850	1/2	39.2699	122.72
11/16	8.44303	5.6727	11/16	17.8678	25.406	3/4	40.0553	127.68
3/4	8.63938	5.9396	3/4	18.0642	25.967	13	40.8407	132.73
13/16	8.83573	6.2126	13/16	18.2605	26.535	1/4	41.6261	137.89
7/8	9.03208	6.4918	7/8	18.4569	27.109	1/2	42.4115	143.14
15/16	9.22843	6.7771	15/16	18.6532	27.688	3/4	43.1969	148.49
3	9.42478	7.0686	6	18.8496	28.274	14	43.9823	153.94
1/16	9.62113	7.3662	1/8	19.2423	29.465	1/4	44.7677	159.48
1/8	9.81748	7.6699	1/4	19.6350	30.680	1/2	45.5531	165.13
3/16	10.0138	7.9798	3/8	20.0277	31.919	3/4	46.3385	170.87
1/4	10.2102	8.2958	1/2	20.4204	33.183	15	47.1239	176.71
5/16	10.4065	8.6179	5/8	20.8131	34.472	1/4	47.9093	182.65
3/8	10.6029	8.9462	3/4	21.2058	35.785	1/2	48.6947	188.69
7/16	10.7992	9.2806	7/8	21.5984	37.122	3/4	49.4801	194.83
1/2	10.9956	9.6211	7	21.9911	38.485	16	50.2655	201.06
9/16	11.1919	9.9678	1/8	22.3838	39.871	1/4	51.0509	207.39
5/8	11.3883	10.321	1/4	22.7765	41.282	1/2	51.8363	213.82
11/16	11.5846	10.680	3/8	23.1692	42.718	3/4	52.6217	220.35
3/4	11.7810	11.045	1/2	23.5619	44.179	17	53.4071	226.98
13/16	11.9773	11.416	5/8	23.9546	45.664	1/4	54.1925	233.71
7/8	12.1737	11.793	3/4	24.3473	47.173	1/2	54.9779	240.53
15/16	12.3700	12.177	7/8	24.7400	48.707	3/4	65.7633	247.45

Diameters, Areas and Circumferences of Circles.—Con.

Diam. Inches.	Circumf. Inches.	Area. Sq. In.	Diam. Inches.	Circumf. Inches.	Area. Sq. In.	Diam. Inches.	Circumf. Inches.	Area. Sq. In.
18	56.5487	254.47	32	100.531	804.25	46	144.513	1661.9
¼	57.3341	261.59	¼	101.316	816.86	¼	145.299	1680.0
½	58.1195	268.80	½	102.102	829.58	½	146.084	1698.2
¾	58.9049	276.12	¾	102.887	842.39	¾	146.869	1716.5
19	59.6903	283.53	33	103.673	855.30	47	147.655	1734.9
¼	60.4757	291.04	¼	104.458	868.31	¼	148.440	1753.5
½	61.2611	298.65	½	105.243	881.41	½	149.226	1772.1
¾	62.0465	306.35	¾	106.029	894.62	¾	150.011	1790.8
20	62.8319	314.16	34	106.814	907.92	48	150.796	1809.6
¼	63.6173	322.06	¼	107.600	921.32	¼	151.582	1828.5
½	64.4026	330.06	½	108.385	934.82	½	152.367	1847.5
¾	66.1880	338.16	¾	109.170	948.42	¾	153.153	1866.5
21	65.9734	346.36	35	109.956	962.11	49	153.938	1885.7
¼	66.7588	354.66	¼	110.741	975.91	¼	154.723	1905.0
½	67.5442	363.05	½	111.527	989.80	½	155.509	1924.2
¾	68.3296	371.54	¾	112.312	1003.8	¾	156.294	1943.9
22	69.1150	380.13	36	113.097	1017.9	50	157.080	1963.5
¼	69.9004	388.82	¼	113.883	1032.1	¼	157.865	1983.2
½	70.6858	397.61	½	114.668	1046.3	½	158.650	2003.0
¾	71.4712	406.49	¾	115.454	1060.7	¾	159.436	2022.8
23	72.2566	415.48	37	116.239	1075.2	51	160.221	2042.8
¼	73.0420	424.56	¼	117.024	1089.8	¼	161.007	2062.9
½	73.8274	433.74	½	117.810	1104.5	½	161.792	2083.1
¾	74.6128	443.01	¾	118.596	1119.2	¾	162.577	2103.3
24	75.3982	452.39	38	119.381	1134.1	52	163.363	2123.7
¼	76.1836	461.86	¼	120.166	1149.1	¼	164.148	2144.2
½	76.9690	471.44	½	120.951	1164.2	½	164.934	2164.8
¾	77.7544	481.11	¾	121.737	1179.3	¾	165.719	2185.4
25	78.5398	490.87	39	122.522	1194.6	53	166.504	2206.2
¼	79.3252	500.74	¼	123.308	1210.0	¼	167.290	2227.0
½	80.1106	510.71	½	124.093	1225.4	½	168.075	2248.0
¾	80.8960	520.77	¾	124.878	1241.0	¾	168.861	2269.1
26	81.6814	530.93	40	125.664	1256.6	54	169.646	2290.2
¼	82.4668	541.19	¼	126.449	1272.4	¼	170.431	2311.5
½	83.2522	551.55	½	127.235	1288.2	½	171.217	2332.8
¾	84.0376	562.00	¾	128.020	1304.2	¾	172.002	2354.3
27	84.8230	572.56	41	128.805	1320.3	55	172.788	2375.8
¼	85.6084	583.21	¼	129.591	1336.4	¼	173.573	2397.5
½	86.3938	593.96	½	130.376	1352.7	½	174.358	2419.2
¾	87.1792	604.81	¾	131.161	1369.0	¾	175.144	2441.1
28	87.9646	615.75	42	131.947	1385.4	56	175.929	2463.0
¼	88.7500	626.80	¼	132.732	1402.0	¼	176.715	2485.0
½	89.5354	637.94	½	133.518	1418.6	½	177.500	2507.2
¾	90.3208	649.18	¾	134.303	1435.4	¾	178.285	2529.4
29	91.1062	660.52	43	135.088	1452.2	57	179.071	2551.8
¼	91.8916	671.96	¼	135.874	1469.1	¼	179.856	2574.2
½	92.6770	683.49	½	136.659	1486.2	½	180.642	2596.7
¾	93.4624	695.13	¾	137.445	1503.3	¾	181.427	2619.4
30	94.2478	706.86	44	138.230	1520.5	58	182.212	2642.1
¼	95.0332	718.69	¼	139.015	1537.9	¼	182.998	2664.9
½	95.8186	730.62	½	139.801	1555.3	½	183.783	2687.8
¾	96.6040	742.64	¾	140.586	1572.8	¾	184.569	2710.9
31	97.3894	754.77	45	141.372	1590.4	59	185.354	2734.0
¼	98.1748	766.99	¼	142.157	1608.2	¼	186.139	2757.2
½	98.9602	779.31	½	142.942	1626.0	½	186.925	2780.5
¾	99.7456	791.73	¾	143.728	1643.9	¾	187.710	2803.9

Diameters, Areas and Circumferences of Circles.—Con.

Diam. Inches.	Circumf. Inches.	Area. Sq. In.	Diam. Inches.	Circumf. Inches.	Area. Sq. In.	Diam. Inches.	Circumf. Inches.	Area. Sq. In.
60	188.496	2827.4	74	232.478	4300.8	88	276.460	6082.1
¼	189.281	2851.0	¼	233.263	4329.9	¼	277.246	6116.7
½	190.066	2874.8	½	234.049	4359.2	½	278.031	6151.4
¾	190.852	2898.6	¾	234.834	4388.5	¾	278.816	6186.2
61	191.637	2922.5	75	235.619	4417.9	89	279.602	6221.1
¼	192.423	2946.5	¼	236.405	4447.4	¼	280.387	6256.1
½	193.208	2970.6	½	237.190	4477.0	½	281.173	6291.2
¾	193.993	2994.8	¾	237.976	4506.7	¾	281.958	6326.4
62	194.779	3019.1	76	238.761	4536.5	90	282.743	6361.7
¼	195.564	3043.5	¼	239.546	4566.4	¼	283.529	6397.1
½	196.350	3068.0	½	240.332	4596.3	½	284.314	6432.6
¾	197.135	3092.6	¾	241.117	4626.4	¾	285.100	6468.2
63	197.920	3117.2	77	241.903	4656.6	91	285.885	6503.9
¼	198.706	3142.0	¼	242.688	4686.9	¼	286.670	6539.7
½	199.491	3166.9	½	243.473	4717.3	½	287.456	6575.5
¾	200.277	3191.9	¾	244.259	4747.8	¾	288.241	6611.5
64	201.062	3217.0	78	245.044	4778.4	92	289.027	6647.6
¼	201.847	3242.2	¼	245.830	4809.0	¼	289.812	6683.8
½	202.633	3267.5	½	246.615	4839.8	½	290.597	6720.1
¾	203.418	3292.8	¾	247.400	4870.7	¾	291.383	6756.4
65	204.204	3318.3	79	248.186	4901.7	93	292.168	6792.9
¼	204.989	3343.9	¼	248.971	4932.7	¼	292.954	6829.5
½	205.774	3369.6	½	249.757	4963.9	½	293.739	6866.1
¾	206.560	3395.3	¾	250.542	4995.2	¾	294.524	6902.9
66	207.345	3421.2	80	251.327	5026.5	94	295.310	6939.8
¼	208.131	3447.2	¼	252.113	5058.0	¼	296.095	6976.7
½	208.916	3473.2	½	252.898	5089.6	½	296.881	7013.8
¾	209.701	3499.4	¾	253.684	5121.2	¾	297.666	7051.0
67	210.487	3525.7	81	254.469	5153.0	95	298.451	7088.2
¼	211.272	3552.0	¼	255.254	5184.9	¼	299.237	7125.6
½	212.058	3578.5	½	256.040	5216.8	½	300.022	7163.0
¾	212.843	3605.0	¾	256.825	5248.9	¾	300.807	7200.6
68	213.628	3631.7	82	257.611	5281.0	96	301.593	7238.2
¼	214.414	3658.4	¼	258.396	5313.3	¼	302.378	7276.0
½	215.199	3685.3	½	259.181	5345.6	½	303.164	7313.8
¾	215.984	3712.2	¾	259.967	5378.1	¾	303.949	7351.8
69	216.770	3739.3	83	260.752	5410.6	97	304.734	7389.8
¼	217.555	3766.4	¼	261.538	5443.3	¼	305.520	7428.0
½	218.341	3793.7	½	262.323	5476.0	½	306.305	7466.2
¾	219.126	3821.0	¾	263.108	5508.8	¾	307.091	7504.5
70	219.911	3848.5	84	263.894	5541.8	98	307.876	7543.0
¼	220.697	3876.0	¼	264.679	5574.8	¼	308.661	7581.5
½	221.482	3903.6	½	265.465	5607.9	½	309.447	7620.1
¾	222.268	3931.4	¾	266.250	5641.2	¾	310.232	7658.9
71	223.053	3959.2	85	267.035	5674.5	99	311.018	7697.7
¼	223.838	3987.1	¼	267.821	5707.9	¼	311.803	7736.6
½	224.624	4015.2	½	268.606	5741.5	½	312.588	7775.6
¾	225.409	4043.3	¾	269.392	5775.1	¾	313.374	7814.8
72	226.195	4071.5	86	270.177	5808.8	100	314.159	7854.0
¼	226.980	4099.8	¼	270.962	5842.6			
½	227.765	4128.2	½	271.748	5876.5			
¾	228.551	4156.8	¾	272.533	5910.6			
73	229.336	4185.4	87	273.319	5944.7			
¼	230.122	4214.1	¼	274.104	5978.9			
½	230.907	4242.9	½	274.889	6013.2			
¾	231.692	4271.8	¾	275.675	6047.6			

TABLE NO. 5.

CUBIC FEET OF AMMONIA GAS PER MINUTE TO PRODUCE ONE TON OF REFRIGERATION PER DAY.

CONDENSER.

		p	103	115	127	139	153	168	185	200	218
	p	t	65°	70°	75°	80°	85°	90°	95°	100	105°
REFRIGERATOR.	4	—20°	5.84	5.9	5.96	6.03	6.06	6.16	6.23	6.30	6.43
	6	—15°	5.35	5.4	5.46	5.52	5.58	5.64	5.70	5.77	5.83
	9	—10°	4.66	4.73	4.76	4.81	4.86	4.91	4.97	5.05	5.08
	13	— 5°	4.09	4.12	4.17	4.21	4.25	4.30	4.35	4.40	4.44
	16	0°	3.59	3.63	3.66	3.70	3.74	3.78	3.83	3.87	3.91
	20	5°	3.20	3.24	3.27	3.30	3.34	3.38	3.41	3.45	3.49
	24	10°	2.87	2.9	2.93	2.96	2.99	3.02	3.06	3.09	3.12
	28	15°	2.59	2.61	2.65	2.68	2.71	2.73	2.76	2.80	2.82
	33	20°	2.31	2.34	2.36	2.38	2.41	2.44	2.46	2.49	2.51
	39	25°	2.06	2.08	2.10	2.12	2.15	2.17	2.20	2.22	2.24
	45	30°	1.85	1.87	1.89	1.91	1.93	1.95	1.97	2.00	2.01
	51	35°	1.70	1.72	1.74	1.76	1.77	1.79	1.81	1.83	1.85

TABLE NO. 6.

PROPERTIES OF SULPHUR DIOXIDE.

t	P	H	h	L	v	w	φ
—22	5.56	157.43	—19.56	176.99	13.17	.076	.4041
—13	7.23	158.64	—16.30	174.95	10.27	.097	.3914
—4	9.27	159.84	—13.05	172.89	8.12	.123	.3791
5	11.76	161.03	—9.79	170.82	6.50	.153	.3673
14	14.74	162.20	—6.53	168.73	5.25	.190	.3559
23	18.31	163.66	—3.27	166.63	4.29	.232	.3449
32	22.53	164.51	0.00	164.51	3.54	.282	.3344
41	27.48	165.65	3.27	162.38	2.93	.340	.3241
50	33.25	166.78	6.55	160.23	2.45	.406	.3142
59	39.33	167.90	9.83	158.07	2.07	.483	.3046
68	47.61	168.99	13.11	155.89	1.75	.570	.2952
77	56.39	170.09	16.39	153.70	1.49	.669	.2861
86	66.36	171.17	19.69	151.49	1.27	.780	.2774
95	77.64	172.24	22.98	149.26	1.09	.906	.2689
104	90.31	173.30	26.28	147.02	.91	1.046	.2607

TABLE NO. 7.

PROPERTIES OF AMMONIA.

t	$\Phi=\frac{L}{T}$	A.	B	t	$\Phi=\frac{L}{T}$	A	B
—40	1.3802	.0000	.0000	60	.9959	.2136	.1707
—35	1.3569	.0118	.0115	65	.9803	.2231	.1768
—30	1.3343	.0236	.0224	70	.9651	.2326	.1825
—25	1.3119	.0351	.0332	75	.9500	.2420	.1881
—20	1.2902	.0465	.0435	80	.9353	.2513	.1936
—15	1.2689	.0578	.0535	85	.9207	.2605	.1990
—10	1.2481	.0690	.0631	90	.9065	.2696	.2042
— 5	1.2276	.0800	.0726	95	.8922	.2787	.2093
0	1.2076	.0910	.0816	100	.8788	.2877	.2140
+5	1.1880	.1018	.0904	105	.8650	.2966	.2186
10	1.1688	.1125	.0989	110	.8516	.3054	.2232
15	1.1500	.1230	.1072	115	.8385	.3141	.2276
20	1.1315	.1335	.1152	120	.8255	.3228	.2319
25	1.1134	.1439	.1229	125	.8129	.3313	.2360
30	1.0957	.1541	.1304	130	.8002	.3398	.2402
35	1.0783	.1643	.1376	135	.7878	.3483	.2441
40	1.0613	.1743	.1446	140	.7756	.3567	.2479
45	1.0445	.1843	.1514	145	.7636	.3650	.2516
50	1.0280	.1941	.1581	150	.7518	.3732	.2552
55	1.0118	.2039	.1645	155	.7402	.3814	.2586

In this table:

$A_1 - A_2 = c \log_e \frac{T_1}{T_1}$; where c is the specific heat of liquid ammonia, c is assumed equal to 1; if any other value is taken the numbers in the table must be multiplied by that value.

$B_1 - B_2 = \varphi_2 - \varphi_1 - \log_e \frac{T_1}{T_2}$.

Examples of use of table:

(1) Per cent. of liquid in wet compression to prevent superheating $= (B_1 - B_2) / \phi_2$.

(2) Superheating above condenser with dry compression $= \frac{2(B_1 - B_2)\,T_1}{1.017 - (B_1 - B_2)}$.

(3) Work to compress, 1 pound of ammonia $= 788\,(T_1 - T_2)\,\varphi_2$;—1 cubic foot $= 788\,(T_1 - T_2)\,\varphi_2\,W_2$.

(4) Mean pressure $= \frac{778}{144}\,(T_1 - T_2)\,\varphi_2\,W_2$.

(5) Equation of adiabatic $A_2 + \varkappa_2\varphi_2 = A_1 + \varkappa_1\varphi_1$.

TABLE NO. 8.

PROPERTIES OF CARBONIC ACID.

t	P	H	h	L	v	w	φ
—22	210	98.35	—37.80	136.15	.4138	2.321	.3108
—13	249	99.14	—32.51	131.65	.3459	2.759	.2945
—4	292	99.88	—26.91	126.79	.2901	3.265	.2785
5	342	100.58	—20.92	121.50	.2438	3.853	.2613
14	396	101.21	—14.49	115.70	.2042	4.535	.2441
23	457	101.81	—7.56	109.37	.1711	5.331	.2262
32	525	102.35	0.00	102.35	.1426	6.265	.2080
41	599	102.84	8.32	94.52	.1177	7.374	.1887
50	680	103.24	17 60	85.64	.0960	8.708	.1679
59	768	103.59	28.22	75.37	.0763	10.356	.1452
68	864	103.84	40.86	62.98	.0577	12.480	.1193
77	968	103 95	57.06	46.89	.0391	15 475	.0873
86	1.080	103 72	84.44	19.28	.0147	21.519	.0353

TABLE NO. 9.

PROPERTIES OF BRINE SOLUTION.

(Common Salt.)

Percentage of Salt by Weight.	Degrees on Salometer at 60° F.	Degrees on Beaume scale.	Specific Gravity at 60° F.	Specific heat.	Weight of one gallon.	Pounds of salt in one gallon.	Weight of one cubic foot.	Freezing point, Degrees F.
0	0	0	1.	1.	8.35	0.	62.4	32.
1	4	1	1.007	0.992	8.4	0.084	62.8	31.8
5	20	5	1.037	0.96	8.65	0.432	64.7	25.4
10	40	10	1.073	0.892	8.95	0.895	66.95	18.6
15	60	15	1.115	0.855	9.3	1.395	69.57	12.2
20	80	19	1.150	0.829	9.6	1.92	71.76	6.86
25	100	23	1.191	0.783	9.94	2.485	74.26	1.00

TABLE NO. 10.

HORSE POWER REQUIRED TO COMPRESS ONE CUBIC FOOT OF AMMONIA PER MINUTE.

CONDENSER PRESSURE AND TEMPERATURE.

REFRIGERATOR PRESSURE & TEMP. p	P	103	115	127	139	153	168	184	200	218
	t	65°	70°	75°	80°	85°	90°	95°	100°	105°
4	−20°	.1809	.1916	.2022	.2128	.2235	.2342	.2448	.2554	.2661
6	−15°	.1864	.1980	.2097	.2214	.2330	.2447	.2563	.2679	.2796
9	−10°	.1937	.2067	.2196	.2325	.2454	.2583	.2712	.2842	.2971
13	−5°	.2001	.2144	.2287	.2430	.2573	.2716	.2859	.3002	.3145
16	0°	.2048	.2206	.2363	.2521	.2679	.2836	.2994	.3151	.3309
20	5°	.2083	.2257	.2430	.2604	.2778	.2952	.3125	.3299	.3473
24	10°	.2096	.2286	.2477	.2667	.2858	.3048	.3239	.3429	.3620
28	15°	.2089	.2298	.2506	.2715	.2924	.3133	.3342	.3551	.3760
33	20°	.2054	.2282	.2510	.2738	.2966	.3195	.3423	.3651	.3879
39	25°	.1992	.2240	.2489	.2738	.2987	.3236	.3485	.3734	.3983
45	30°	.1897	.2169	.2440	.2711	.2982	.3253	.3524	.3795	.4066
51	35°	.1768	.2062	.2357	.2651	.2946	.3241	.3535	.3830	.4124

TABLE NO. 11.

MEAN PRESSURE OF DIAGRAM OF AMMONIA COMPRESSOR.

CONDENSER PRESSURE AND TEMPERATURE.

REFRIGERATOR PRESSURE & TEMP.	p	103	115	127	139	153	168	184	200	218
p	t°	65°	70°	75°	80°	85°	90°	95°	100°	105°
4	−20°	41.46	43.91	46.34	48.77	51.23	53.68	56.11	58.54	60.99
6	−15°	42.72	45.38	47.90	50.74	53.40	56.08	58.86	61.40	64.08
9	−10°	44.40	47.38	50.33	53.29	56.25	59.20	62.16	65.14	68.09
13	− 5°	45.86	49.15	52.42	55.70	58.97	62.25	65.53	68.81	72.08
16	0°	46.94	50.56	54.16	57 78	61.40	65.00	68.62	72.22	75.84
20	5°	47.74	51.73	55.70	59.68	63.67	67.66	71.62	75.61	79.61
24	10°	48.04	52.40	56.77	61.13	65.51	69.86	74.24	78.59	82.97
28	15°	47.88	52.67	57.44	62.23	67.02	71.81	76.60	81.39	86.18
33	20°	47.08	52.30	57.53	62.75	67.98	73.23	78.46	83.68	88.91
39	25°	45.06	51.34	57.05	62.75	68.46	74.17	79.88	85.58	91.29
45	30°	43.16	49.71	55.92	62.14	68.35	74.56	80.77	86 98	93.19
51	35°	40.52	47.26	54.02	60.76	67.52	74.28	81.02	87.78	94.52

TABLE NO. 12.

Reaumur, Fahrenheit, and Celsius Thermometers Compared.

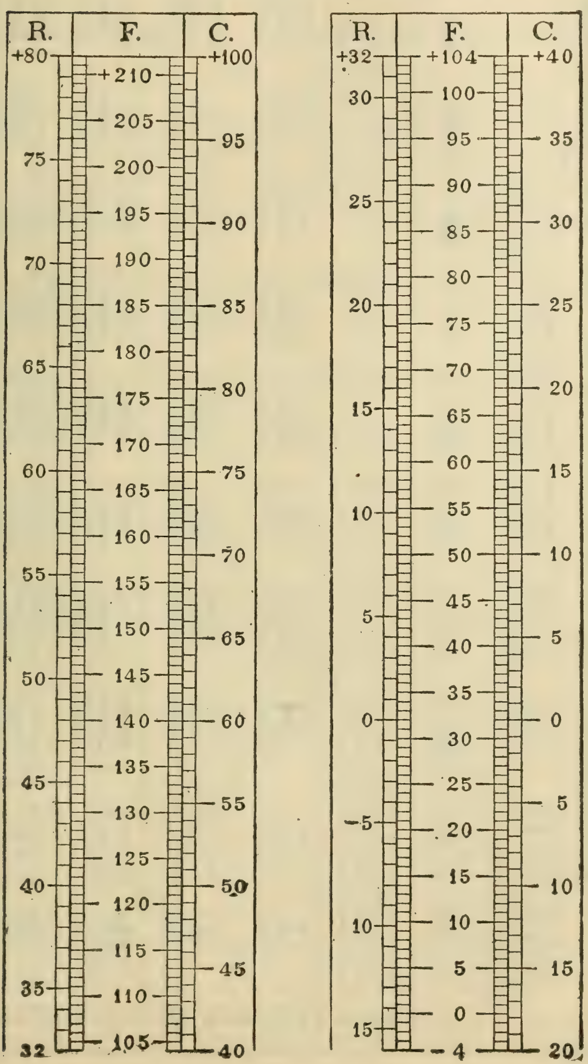


TABLE NO. 13.

HORSE POWER REQUIRED TO PRODUCE ONE TON OF REFRIGERATION.

CONDENSER PRESSURE AND TEMPERATURE.

REFRIGERATOR PRESSURE & TEMP.

	p	103	115	127	139	153	168	184	200	218
p	t	65°	70°	75°	80°	85°	90°	95°	100°	105°
4	— 20°	1.0584	1.1304	1.2051	1.2832	1.3611	1.4427	1.5251	1.6090	1.6910
6	— 15°	.9972	1.0692	1.1450	1.2221	1.3001	1.4101	1.4609	1.5458	1.7300
9	— 10°	.9026	.9777	1.0453	1.1183	1.1926	1.2602	1.3471	1.4352	1.5093
13	— 5°	.8184	.8833	.9537	1.0230	1.0935	1.1679	1.2437	1.3209	1.3964
16	0°	.7352	.8008	.8648	.9328	1.0019	1.0718	1.1467	1.2194	1.2547
20	5°	.6665	7312	.7946	.8593	.9278	.9978	1.0656	1.1381	1.2121
24	10°	.5915	.6629	.7257	.7894	.8545	.9205	.9911	1.0595	1.1294
28	15°	.5410	.5998	.6641	.7276	.7924	.8553	.9224	.9943	1.0603
33	20°	.4745	.5340	.5923	.6716	.7148	.7796	.8420	.9031	.9736
39	25°	.4103	.4659	.5227	.5804	.5992	.7022	.7667	.8289	.8922
45	30°	.3509	.4056	.4612	.5178	.5755	.6353	.6944	.7590	.8172
51	35°	.3005	.3546	.4101	.4666	.5214	.5804	.6398	.7009	7629

TABLE NO. 14.

Weight of Rivets and Round Headed Bolts Without Nuts, Per 100.

LENGTH FROM UNDER HEAD. ONE CUBIC FOOT WEIGHING 480 LBS.

LENGTH INCHES.	3/8″ DIAM.	1/2″ DIAM.	5/8″ DIAM.	3/4″ DIAM.	7/8″ DIAM.	1″ DIAM.	1 1/8″ DIAM.	1 1/4″ DIAM.
1 1/4	5.4	12.6	21.5	28.7	43.1	65.3	91.5	123.
1 1/2	6.2	13.9	23.7	31.8	47.3	70 7	98.4	133.
1 3/4	6.9	15.3	25.8	34.9	51.4	76.2	105.	142.
2	7.7	16.6	27.9	37.9	55.6	81.6	112.	150.
2 1/4	8.5	18.0	30.0	41.0	59.8	87.1	119.	159.
2 1/2	9.2	19.4	32.2	44.1	63.0	92.5	126.	167.
2 3/4	10.0	20.7	34.3	47.1	68.1	98.0	133.	176.
3	10.8	22.1	36.4	50.2	72.3	103.	140.	184.
3 1/4	11.5	23.5	38.6	53.3	76.5	109.	147.	193.
3 1/2	12.3	24.8	40.7	56.4	80.7	114.	154.	201.
3 3/4	13.1	26.2	42.8	59.4	84.8	120.	161.	210.
4	13.8	27.5	45.0	62.5	89.0	125.	167.	218.
4 1/4	14.6	28.9	47.1	65.6	93.2	131.	174.	227.
4 1/2	15.4	30.3	49.2	68.6	97.4	136.	181.	236.
4 3/4	16.2	31.6	51.4	71.7	102.	142.	188.	244.
5	16.9	33.0	53.5	74.8	106.	147.	195.	253.
5 1/4	17.7	34.4	55.6	77.8	110.	153.	202.	261.
5 1/2	18.4	35.7	57.7	80.9	114.	158.	209.	270.
5 3/4	19.2	37.1	59.9	84.0	118.	163.	216.	278.
6	20.0	38.5	62.0	87.0	122.	169.	223.	287.
6 1/2	21.5	41.2	66.3	93.2	131.	180.	236.	304.
7	23.0	43.9	70.5	99.3	139.	191.	250.	321.
7 1/2	24.6	46.6	74.8	106.	147.	202.	264.	338.
8	26.1	49.4	79.0	112.	156.	213.	278.	355.
8 1/2	27.6	52.1	83.3	118.	164.	223.	292.	372.
9	29.2	54.8	87.6	124.	173.	234.	306.	389.
9 1/2	30.7	57.6	91.8	130.	181.	245.	319.	406.
10	32.2	60.3	96.1	136.	189.	256.	333.	423.
10 1/2	33.8	63.0	101.	142.	198.	267.	347.	440.
11	35.3	65.7	105.	148.	206.	278.	361.	457.
11 1/2	36.8	68.5	109.	155.	214.	289.	375.	474.
12	38.4	71.2	113.	161.	223.	300.	388.	491.
Heads.	1.8	5.7	10.9	13.4	22.2	38.0	57.0	82.0

TABLE NO. 15.

Weight and Strength of Iron Bolts.

Ends enlarged or upset.			Ends not enlarged.		Ends enlarged or upset.			Ends not enlarged.	
Diameter of shank.	Weight per foot run.	Breaking strain.	Diameter of shank.	Weight per foot run.	Diameter of shank.	Weight per foot run.	Breaking strain.	Diameter of shank.	Weight per foot run.
Ins.	Pounds.	Tons. 2240 lbs.	Ins.	Lbs.	Ins.	Lbs.	Tons 2240 lbs.	Ins.	Lbs.
1/8	.0414	.245	....	....	1 3/4	8.10	45.7	2.14	12.0
3/16	.093	.553	....	....	1 13/16	8.69	49.0	2.22	12.9
1/4	.165	.983	.35	.321	1 7/8	9.30	52.5	2.30	13.8
5/16	.258	1.53	.43	.452	1 15/16	9.93	56.0	2.38	14.7
3/8	.372	2.21	.50	.654	2	10.6	59.7	2.45	15.7
7/16	.506	3.00	.58	.897	2 1/8	12.0	63.8	2.59	17.5
1/2	.661	3.93	.66	1.14	2 1/4	13.4	71.6	2.73	19.5
9/16	.837	4.97	.73	1.41	2 3/8	14.9	79.7	2.88	21.6
5/8	1.03	6.14	.80	1.67	2 1/2	16.5	88.4	3.02	23.9
11/16	1.25	7.42	.88	2.03	2 5/8	18.2	97.4	3.16	26.1
3/4	1.49	8.83	.96	2.41	2 3/4	20.0	106.9	3.30	28.5
13/16	1.75	10.4	1.04	2.81	2 7/8	21.9	116.8	3.45	31.1
7/8	2.03	12.0	1.12	3.26	3	23.8	127.2	3.60	33.9
15/16	2.33	13.8	1.20	3.77	3 1/4	27.9	141.0	3.86	39.1
1	2.65	15.7	1.27	4.27	3 1/2	32.4	163.6	4.12	44.4
1 1/16	2.99	16.8	1.35	4.77	3 3/4	37.2	187.7	4.41	51.0
1 1/8	3.35	18.9	1.42	5.28	4	42.3	213.6	4.70	57.8
1 3/16	3.73	21.1	1.49	5.81	4 1/4	47.8	227.0	4.98	65.2
1 1/4	4.13	23.3	1.55	6.39	4 1/2	53.6	254.5	5.25	72.9
1 5/16	4.56	25.7	1.64	7.04	4 3/4	59.7	283.5	5.53	80.5
1 3/8	5.00	28.2	1.72	7.74	5	66.1	314.2	5.80	88.1
1 7/16	5.47	30.8	1.80	8.48	5 1/4	72.9	324.7	6.08	97.0
1 1/2	5.95	33.6	1.87	9.20	5 1/2	80.0	356.4	6.36	106.
1 9/16	6.46	36.4	1.94	9.88	5 3/4	87.5	389.5	6.63	116.
1 5/8	6.99	39.4	2.00	10.6	6	95.2	424.1	6.90	126.
1 11/16	7.53	42.5	2.07	11.3	....	....	....	....	....

For *square bars* increase the breaking strains 1/4 part.

A long upset rod is no stronger than one not upset, against ***slowly applied*** loads or strains. **Therefore in such cases the column of greatest diameter in the table should be used.**

TABLE NO. 16.

Boiling Points of Various Substances.

At Atmospheric Pressure at Sea Level.

Substance.	Degrees Fahr.	Substance.	Degrees Fahr.
Alcohol	173	Sulphur	570
Ammonia	140	Sulphuric Acid, s. g 1.848	590
Benzine	176	Sulphuric Acid, s. g. 1.3	240
Coal Tar	325	Sulphuric Ether	100
Linseed Oil	597	Turpentine	315
Mercury	648	Water	212
Naptha	186	Water, Sea	213.2
Nitric Acid, s. g. 1.42	248	Water, Saturated Brine	226
Nitric Acid, s. g. 1.5	210	Wood Spirit	150
Petroleum Rectified	316		

TABLE NO. 17.

Melting Points of Metals.

From D. K. C.

Metal.	Degrees Fahr.
Aluminum	Full Red Heat.
Antimony	1150
Bismuth	507
Bronze	1690
Copper	1996
Gold, Standard	2156
Gold, Pure	2282
Iron, Cast, Gray	2012
Iron, Cast, White	1922 to 2012
Iron, Wrought	2912
Lead	617
Mercury	—39
Silver	1873
Steel	2372 to 2552
Tin	442
Zinc	773

Melting Points of Various Solids.

From D. K. C. and H.

Substance.	Degrees Fahr.
Carbonic Acid	—108
Glass	2377
Ice	32
Lard	95
Nitro-Glycerine	45
Phosphorus	112
Pitch	91
Saltpetre	606
Spermacetti	120
Stearine	109 to 120
Sulphur	239
Tallow	92
Turpentine	14
Wax, Rough	142
Wax, Bleached	154

Melting Points of Fusible Plugs.

From D. K. C.

	Softens at	Melts at		Softens at	Melts at
2 Tin, 2 Lead	365	372	2 Tin, 7 Lead	377½	388
Tin, 6 Lead	372	383	2 Tin, 8 Lead	395¼	408

TABLE NO. 18.

Showing Coal Consumption Per Square Foot of Grate Surface Per Hour for Different Areas of Grate.

Sq. Feet of Grate Surface.	Pounds of Fuel Burned Per Hour.										
	50	100	200	300	400	500	600	700	800	900	1000
4	12.50	25.00	50.00	75.00	100.00	125.00	150.00	175.00	200.00	225.00	250.00
6	8.33	16.60	33.30	50.00	66.60	83.30	100.00	116.60	133.30	150.00	166.60
8	6.25	12.50	25.00	37.50	50.00	62.50	75.00	87.50	100.00	112.50	125.00
10	5.00	10.00	20.60	30.00	40.00	50.00	60.00	70.00	80.00	90.00	100.00
12	4.16	8.33	16.60	25.00	33.30	41.66	50.00	58.33	66.60	75.00	83.32
14	3.57	7.14	14.28	21.40	28.57	35.71	42.80	50.00	57.14	60.90	71.42
16	3.12	6.25	12.50	18.70	25.10	31.25	37.40	43.75	50.20	53.25	62.50
18	2.77	5.55	11.11	16.60	22.22	27.70	33.20	38.80	44.44	50.00	55.40
20	2.50	5.00	10.00	15.00	20.00	25.00	30.00	35.00	40.00	45.00	50.00
22	2.27	4.55	9.11	13.68	18.22	22.72	27.36	31.81	36.44	40.90	45.44
24	2.08	4.16	8.33	12.50	16.66	20.83	25.00	28.33	33.32	37.50	41.66
26	1.92	3.84	7.69	11.53	15.38	19.23	23.06	26.92	30.76	34.61	38.46
28	1.78	3.57	7.14	10.71	14.28	17.85	21.42	25.00	28.56	32.14	35.60
30	1.66	3.33	6.66	10.00	13.33	16.66	20.00	23.33	26.66	30.00	33.32
32	1.56	3.12	6.25	9.37	12.50	15.62	18.74	21.87	25.00	28.12	31.25
34	1.47	2.94	5.88	8.82	11.77	14.70	17.64	20.58	23.54	26.47	29.40
36	1.38	2.77	5.55	8.33	11.11	13.88	16.66	19.44	22.22	25.00	27.76
38	1.31	2.63	5.25	7.89	10.50	13.15	15.78	18.42	21.00	23.68	26.30
40	1.25	2.50	5.00	7.50	10.00	12.50	15.00	17.50	20.00	22.50	25.00
42	1.19	2.38	4.73	7.14	9.52	11.90	14.28	16.66	19.04	21.42	23.80
44	1.13	2.27	4.54	6.81	9.09	11.36	13.62	15.99	18.18	20.45	22.72
46	1.08	2.17	4.34	6.52	8.69	10.86	13.04	15.21	17.38	19.56	21.72
48	1.04	2.08	4.16	6.29	8.33	10.41	12.58	14.66	16.66	18.75	20.82
50	1.00	2.00	4.00	6.00	8.00	10.00	12.00	14.00	16.00	18.00	20.00

TABLE NO. 19.

Showing Cost of Coal Per Annum, (365 days). For 300 days.

Tons Per Day.	Price Per Ton. $1.00	$1.25	$1.50	$1.75	$2.00	$2.25	$2.50	$2.75	$3.00	$3.25	$3.50
½	182.5	228.12	273.75	319.37	365	411.60	456.24	501.80	548.5	593.10	638.75
1	365	456.25	547.50	638.75	730	821.25	912.50	1003.75	1095	1186.25	1277.50
2	730	912.50	1095.00	1277.50	1460	1642.50	1825.00	2007.50	2190	2372.50	2555.00
3	1095	1368.75	1642.50	1916.25	2190	2463.75	2737.50	3011.25	3285	3558.75	3832.50
4	1460	1825.00	2190.00	2555.00	2920	3285.00	3650.00	4015.00	4380	4745.00	5110.00
5	1825	2281.25	2737.50	3193.75	3650	4106.25	4562.50	5018.75	5475	5931.25	6387.50
6	2190	2737.50	3285.00	3832.50	4380	4927.50	5475.00	6022.50	6570	7117.50	7665.00
7	2555	3193.75	3832.50	4471.25	5110	5748.75	6397.50	7026.25	7665	8303.75	8942.50
8	2920	3650.00	4380.00	5110.00	5840	6570.00	7300.00	8030.00	8760	9490.09	10220.00
9	3295	4118.75	4942.50	5766.25	6590	7413.75	8237.50	9061.25	9895	10698.75	11532.50
10	3650	4562.50	5475.00	6387.50	7300	8212.50	9125.00	10036.00	10950	11862.50	12775.00
11	4015	5018.75	6022.50	7026.25	8030	9033.75	10037.50	11041.25	12045	13048.75	14052.50
12	4380	5475.00	6560.00	7665.00	8760	9855.00	10950.00	12045.00	13140	14235.00	15330.00
13	4745	5941.25	7117.50	8503.75	9490	10676.75	11882.50	13048.75	14235	15421.25	17006.50
14	5110	6387.50	7665.00	8942.50	10230	11497.50	12775.00	14052.50	14330	16607.50	17885.00
15	5475	6843.75	8212.50	9581.25	10950	12318.75	13687.50	15056.25	16425	17793.75	19162.50
16	5840	7300.00	8760.00	10220.00	11680	13140.00	14600.00	16060.00	17520	18980.00	20440.00
17	6205	7756.25	9307.50	10858.75	12410	13961.25	15512.50	17063.75	18615	20166.25	21716.50
18	6590	8237.50	9835.00	11532.50	13180	14827.50	16475.00	18122.50	19770	21417.50	23065.00
19	6945	8681.25	10417.50	12153.75	13890	15626.25	17362.50	19098.75	20835	22571.25	24306.50
20	7300	9125.00	10950.00	12775.00	14600	16425.00	18250.00	20075.00	21900	23725.00	25550.00
21	7665	9581.25	11497.50	13413.75	15330	17246.25	19162.50	21078.85	22995	24911.25	26827.50
22	8030	10037.50	12045.00	14052.50	16060	18067.50	20075.00	22082.50	24090	26097.50	28105.00
23	8395	10493.75	12592.50	14691.25	16790	18888.75	20937.50	23086.25	25185	27283.75	29382.50
24	8760	10950.00	13140.00	15330.00	17520	0710.00	21900.00	24090.00	26280	28470.00	30660.00

TABLE NO. 20.

Capacity of Reservoirs in Gallons.

NOTE — The columns headed Length and Width denote the length and width in feet; the columns headed Gallons denote the capacity in U. S. gallons for one foot in depth.

Length and Width	Gallons.	Length and Width.	Gallons.	Length and Width.	Gallons.	Length and Width.	Gallons.
1 x 1	7.481	15 x 7	785.455	23 x 10	1720.519	13 x 13	1264.208
2 x 1	14.961	16 x 7	837.818	24 x 10	1795.325	14 x 13	1361.454
3 x 1	22.442	17 x 7	890.182	25 x 10	1870.130	15 x 13	1458.701
2 x 2	29.922	18 x 7	942.545	26 x 10	1944.935	16 x 13	1555.948
3 x 2	44.883	19 x 7	994.909	27 x 10	2019.740	17 x 13	1653.195
4 x 2	59.844	20 x 7	1047.273	28 x 10	2094.545	18 x 13	1750.442
5 x 2	74.805	21 x 7	1099.636	29 x 10	2169.351	19 x 13	1847.688
6 x 2	89.766	8 x 8	478.753	30 x 10	2244.156	20 x 13	1944.935
3 x 3	67.325	9 x 8	538.597	11 x 11	905.143	21 x 13	2042.182
4 x 3	89.766	10 x 8	598.442	12 x 11	987.429	22 x 13	2139.429
5 x 3	112.208	11 x 8	658.286	13 x 11	1069.714	23 x 13	2236.675
6 x 3	134.649	12 x 8	718.130	14 x 11	1152.000	24 x 13	2333.922
7 x 3	157.091	13 x 8	777.974	15 x 11	1234.286	25 x 13	2431.169
8 x 3	179.532	14 x 8	837.818	16 x 11	1316.571	26 x 13	2528.416
9 x 3	201.974	15 x 8	897.662	17 x 11	1398.857	27 x 13	2625.662
4 x 4	119.688	16 x 8	957.507	18 x 11	1481.143	28 x 13	2722.909
5 x 4	149.610	17 x 8	1017.351	19 x 11	1563.429	29 x 13	2820.156
6 x 4	179.532	18 x 8	1077.195	20 x 11	1645.714	30 x 13	2917.403
7 x 4	209.455	19 x 8	1137.039	21 x 11	1728.000	31 x 13	3014.649
8 x 4	239.377	20 x 8	1196.883	22 x 11	1810.286	32 x 13	3111.896
9 x 4	269.299	21 x 8	1256.727	23 x 11	1892.571	33 x 13	3209.143
10 x 4	299.221	22 x 8	1316.571	24 x 11	1974.857	34 x 13	3306.390
11 x 4	329.143	23 x 8	1376.416	25 x 11	2057.143	35 x 13	3403.636
12 x 4	359.065	24 x 8	1436.260	26 x 11	2139.428	36 x 13	3500.883
5 x 5	187.013	9 x 9	605.922	27 x 11	2221.714	37 x 13	3598.130
6 x 5	224.416	10 x 9	673.247	28 x 11	2304.000	38 x 13	3695.377
7 x 5	261.818	11 x 9	740.571	29 x 11	2386.286	39 x 13	3792.623
8 x 5	299.221	12 x 9	807.896	30 x 11	2468.571	14 x 14	1466.182
9 x 5	336.623	13 x 9	875.221	31 x 11	2550.857	15 x 14	1570.909
10 x 5	374.026	14 x 9	942.545	32 x 11	2633.143	16 x 14	1675.636
11 x 5	411.429	15 x 9	1009.870	33 x 11	2715.429	17 x 14	1780.363
12 x 5	448.831	16 x 9	1077.195	12 x 12	1077.195	18 x 14	1885.091
13 x 5	486.234	17 x 9	1144.519	13 x 12	1166.961	19 x 14	1989.818
14 x 5	523.636	18 x 9	1211.844	14 x 12	1256.727	20 x 14	2094.545
15 x 5	561.039	19 x 9	1279.169	15 x 12	1346.493	21 x 14	2199.263
6 x 6	269.299	20 x 9	1346.493	16 x 12	1436.260	22 x 14	2304.000
7 x 6	314.182	21 x 9	1413.818	17 x 12	1526.026	23 x 14	2408.727
8 x 6	359.065	22 x 9	1481.143	18 x 12	1615.792	24 x 14	2513.454
9 x 6	403.948	23 x 9	1548.467	19 x 12	1705.558	25 x 14	2618.182
10 x 6	448.831	24 x 9	1615.792	20 x 12	1795.325	26 x 14	2722.909
11 x 6	493.714	25 x 9	1683.117	21 x 12	1885.091	27 x 14	2827.636
12 x 6	538.597	26 x 9	1750.442	22 x 12	1974.857	28 x 14	2932.364
13 x 6	583.480	27 x 9	1817.766	23 x 12	2064.623	29 x 14	3037.091
14 x 6	628.364	10 x 10	748.052	24 x 12	2154.390	30 x 14	3141.818
15 x 6	673.247	11 x 10	822.857	25 x 12	2244.156	31 x 14	3246.545
16 x 6	718.130	12 x 10	897.662	26 x 12	2333.922	32 x 14	3351.273
17 x 6	763.013	13 x 10	972.467	27 x 12	2423.688	33 x 14	3456.000
18 x 6	807.896	14 x 10	1047.273	28 x 12	2513.455	34 x 14	3560.727
7 x 7	366.545	15 x 10	1122.078	29 x 12	2603.221	35 x 14	3665.454
8 x 7	418.909	16 x 10	1196.883	30 x 12	2692.987	36 x 14	3770.182
9 x 7	471.273	17 x 10	1271.688	31 x 12	2782.753	37 x 14	3874.909
10 x 7	523.636	18 x 10	1346.493	32 x 12	2872.520	38 x 14	3979.636
11 x 7	576.000	19 x 10	1421.299	33 x 12	2962.286	39 x 14	4084.364
12 x 7	628.364	20 x 10	1496.104	34 x 12	3052.052	40 x 14	4189.091
13 x 7	680.727	21 x 10	1570.909	35 x 12	3141.818	41 x 14	4293.818
14 x 7	733.091	22 x 10	1645.714	36 x 12	3231.585	42 x 14	4398.545

TABLE NO. 21.

Capacity of Reservoirs in Gallons. — *Continued.*

Length and Width	Gallons.	Length and Width.	Gallons.	Length and Width	Gallons	Length and Width.	Gallons.
15 x 15	1683.117	28 x 17	3560.727	33 x 20	4937.143	52 x 28	10891.636
16 x 15	1795.325	29 x 17	3687.896	34 x 20	5086.753	54 x 28	11310.545
17 x 15	1907.532	30 x 17	3815.065	35 x 20	5236.364	56 x 28	11729.454
18 x 15	2019.740	31 x 17	3942.234	36 x 20	5385.974	30 x 30	6732.467
19 x 15	2131.948	32 x 17	4069.403	37 x 20	5535.584	32 x 30	7181.299
20 x 15	2244.156	33 x 17	4196.571	38 x 20	5685.195	34 x 30	7630.130
21 x 15	2356.364	34 x 17	4323.740	39 x 20	5834.805	36 x 30	8078.961
22 x 15	2468.571	18 x 18	2423.688	40 x 20	5984.416	38 x 30	8527.792
23 x 15	2580.779	19 x 18	2558.338	22 x 22	3620.571	40 x 30	8976.623
24 x 15	2692.987	20 x 18	2692.987	24 x 22	3949.714	42 x 30	9425.454
25 x 15	2805.195	21 x 18	2827.636	26 x 22	4278.857	44 x 30	9874.286
26 x 15	2917.403	22 x 18	2962.286	28 x 22	4608.000	46 x 30	10323.117
27 x 15	3029.610	23 x 18	3096.935	30 x 22	4937.143	48 x 30	10771.948
28 x 15	3141.818	24 x 18	3231.584	32 x 22	5266.286	50 x 30	11220.779
29 x 15	3254.026	25 x 18	3366.234	34 x 22	5595.429	52 x 30	11669.610
30 x 15	3366.234	26 x 18	3500.883	36 x 22	5924.571	54 x 30	12118.442
31 x 15	3478.442	27 x 18	3635.532	38 x 22	6253.714	56 x 30	12567.273
32 x 15	3590.649	28 x 18	3770.182	40 x 22	6582.857	58 x 30	13016.104
33 x 15	3702.857	29 x 18	3904.831	42 x 22	6912.000	60 x 30	13464.935
34 x 15	3815.065	30 x 18	4039.480	44 x 22	7241.143	32 x 32	7660.052
35 x 15	3927.273	31 x 18	4174.130	24 x 24	4308.779	34 x 32	8138.805
36 x 15	4039.480	32 x 18	4308.779	26 x 24	4667.844	36 x 32	8617.558
37 x 15	4151.688	33 x 18	4443.429	28 x 24	5026.909	38 x 32	9096.312
38 x 15	4263.896	34 x 18	4578.078	30 x 24	5385.974	40 x 32	9575.065
39 x 15	4376.104	35 x 18	4712.727	32 x 24	5745.039	42 x 32	10053.818
40 x 15	4488.312	36 x 18	4847.377	34 x 24	6104.104	44 x 32	10532.571
41 x 15	4600.519	19 x 19	2700.467	36 x 24	6463.169	46 x 32	11011.325
42 x 15	4712.727	20 x 19	2842.597	38 x 24	6822.234	48 x 32	11490.078
43 x 15	4824.935	21 x 19	2984.727	40 x 24	7181.299	50 x 32	11968.831
44 x 15	4937.143	22 x 19	3126.857	42 x 24	7540.364	52 x 32	12447.584
45 x 15	5049.351	23 x 19	3268.987	44 x 24	7899.429	54 x 32	12926.338
16 x 16	1915.013	24 x 19	3411.117	46 x 24	8258.493	56 x 32	13405.091
17 x 16	2034.701	25 x 19	3553.247	48 x 24	8617.558	58 x 32	13883.844
18 x 16	2154.390	26 x 19	3695.377	26 x 26	5056.831	60 x 32	14362.597
19 x 16	2274.078	27 x 19	3837.506	28 x 26	5445.818	62 x 32	14841.351
20 x 16	2393.766	28 x 19	3979.636	30 x 26	5834.805	64 x 32	15320.104
21 x 16	2513.454	29 x 19	4121.766	32 x 26	6223.792	34 x 34	8647.480
22 x 16	2633.143	30 x 19	4263.896	34 x 26	6612.779	36 x 34	9156.156
23 x 16	2752.831	31 x 19	4406.026	36 x 26	7001.766	38 x 34	9664.831
24 x 16	2872.519	32 x 19	4548.156	38 x 26	7390.753	40 x 34	10173.506
25 x 16	2992.208	33 x 19	4690.286	40 x 26	7779.740	42 x 34	10682.182
26 x 16	3111.896	34 x 19	4832.416	42 x 26	8168.727	44 x 34	11190.857
27 x 16	3231.584	35 x 19	4974.545	44 x 26	8557.714	46 x 34	11699.532
28 x 16	3351.273	36 x 19	5116.675	46 x 26	8946.701	48 x 34	12208.208
29 x 16	3470.961	37 x 19	5258.805	48 x 26	9835.688	50 x 34	12716.883
30 x 16	3590.649	38 x 19	5400.935	50 x 26	9724.675	52 x 34	13225.558
31 x 16	3710.338	20 x 20	2992.208	52 x 26	10113.662	54 x 34	13734.234
32 x 16	3830.026	21 x 20	3141.818	28 x 28	5864.727	56 x 34	14242.909
17 x 17	2161.870	22 x 20	3291.429	30 x 28	6283.636	58 x 34	14751.584
18 x 17	2289.039	23 x 20	3441.039	32 x 28	6702.545	60 x 34	15260.260
19 x 17	2416.208	24 x 20	3590.649	34 x 28	7121.454	62 x 34	15768.935
20 x 17	2543.377	25 x 20	3740.260	36 x 28	7540.364	64 x 34	16277.610
21 x 17	2670.545	26 x 20	3889.870	38 x 28	7959.273	66 x 34	16786.286
22 x 17	2797.714	27 x 20	4039.480	40 x 28	8378.182	68 x 34	17294.961
23 x 17	2924.883	28 x 20	4189.091	42 x 28	8797.091	36 x 36	9694.753
24 x 17	3052.052	29 x 20	4338.701	44 x 28	9216.000	38 x 36	10233.351
25 x 17	3179.221	30 x 20	4488.312	46 x 28	9634.909	40 x 36	10771.948
26 x 17	3306.390	31 x 20	4637.922	48 x 28	10053.818	42 x 36	11310.545
27 x 17	3433.558	32 x 20	4787.532	50 x 28	10472.727	44 x 36	11849.143

The United States inspection laws allow 20 per cent more pressure to be carried on boilers with double riveted longitudinal seams, than on single riveted boilers.

TABLE NO. 22.

Table of Piston Speeds in Feet Per Minute

Stroke in Inches.	REVOLUTIONS PER MINUTE.																								Stroke in Inches.
	50	60	75	80	90	100	120	125	140	150	160	170	180	200	225	240	250	275	300	320	325	350	360	375	400	
6	50	60	75	80	90	100	120	125	140	150	160	170	180	200	225	240	250	275	300	320	325	350	360	375	400	6
7	58	70	87	93	105	117	140	146	163	175	187	198	210	233	262	280	292	321	350	373	379	408	420	437	467	7
8	67	80	100	107	120	133	160	167	187	200	213	227	240	267	300	320	333	367	400	427	433	467	480	500	533	8
9	75	90	112	120	135	150	180	187	210	225	240	255	270	300	337	360	375	412	450	480	487	525	540	562	600	9
10	83	100	125	133	150	167	200	208	233	250	267	283	300	333	375	400	417	458	500	533	542	583	600	625	667	10
11	92	110	137	147	165	183	220	229	257	275	293	312	330	367	412	440	458	504	550	587	596	642	660	687	733	11
12	100	120	150	160	180	200	240	250	280	300	320	340	360	400	450	480	500	550	600	640	650	700	720	750	800	12
14	117	140	175	187	210	233	280	292	303	350	373	397	420	466	525	560	583	642	700	747	758	817	840	875	933	14
15	125	150	187	200	225	250	300	312	350	375	400	425	450	500	562	600	625	687	750	800	812	875	900	937	1000	15
16	133	160	200	213	240	267	320	333	373	400	427	453	480	533	600	640	667	733	800	853	867	933	960	1000	1067	16
18	150	180	225	240	270	300	360	375	420	450	480	510	540	600	675	720	750	825	900	960	975	1050	1080	1125	1200	18
20	167	200	250	267	300	334	400	417	467	500	533	587	600	667	750	800	833	917	1000	1067	1083	1167	1200	1250	1333	20
22	193	220	275	293	330	367	440	458	513	550	587	623	660	733	825	880	917	1008	1100	1173	1192	1233	1320	1375	1467	22
24	200	240	300	320	360	400	480	500	560	600	640	680	720	800	900	960	1000	1100	1200	1280	1300	1400	1440	1500	1600	24
26	217	260	325	347	390	433	520	542	607	650	693	733	780	867	975	1040	1083	1192	1300	1360	1408	1517	1560	1625	1731	26
28	233	280	350	373	420	467	560	583	653	700	746	793	840	933	1050	1120	1166	1283	1400	1494	1516	1634	1680	1750	1866	28
30	250	300	375	400	450	500	600	625	700	750	800	850	900	1000	1125	1200	1250	1375	1500	1600	1624	1750	1800	1874	2000	30
36	300	360	450	480	540	600	720	750	840	900	960	1020	1080	1200	1350	1440	1500	1650	1800	1920	1950	2100	2160	2250	2400	36
40	333	400	500	533	600	667	800	833	933	1000	1066	1133	1200	1333	1500	1600	1666	1834	2000	2134	2166	2334	2400	2500	2666	40
42	350	420	525	560	630	700	840	875	980	1050	1120	1190	1360	1400	1575	1680	1749	1926	2100	2241	2274	2451	2520	2625	2799	42
48	400	480	600	640	720	800	960	1000	1120	1200	1300	1260	1440	1600	1800	1920	2000	2200	2400	2560	2600	2800	2880	3000	3200	48
54	450	540	675	720	810	900	1080	1125	1260	1350	1440	1530	1620	1800	2025	2160	2250	2475	2700	2880	2925	3150	3240	3375	3600	54
60	500	600	750	800	900	1000	1200	1250	1400	1500	1600	1700	1800	2000	2250	2400	2500	2750	3000	3200	3248	3500	3600	3748	4000	60
66	550	660	825	880	990	1100	1320	1375	1540	1650	1761	1869	1980	2199	2475	2640	2751	3024	3300	3519	3576	3699	3960	4125	4401	66
72	600	720	900	960	1080	1200	1440	1500	1680	1800	1920	2040	2160	2400	2700	2880	3000	3300	3600	3840	3900	4200	4320	4500	4800	72

TABLE NO. 23.

Properties of Saturated Steam.

Pressure per Square Inch.	Pressure Above Zero.	Temperature.	Latent Heat.	Total Heat Above Zero.	Weight of One CubicFoot.
Inches of Mercury.	Pounds per Square Inch.	Fahr. Deg.	B. T. U.	Fahr., B. T. U.	Pounds.
2.04	1	102.00	1042 96	1145 05	0030
4 07	2	126 26	1026 01	1152 45	.0058
6.11	3	141.62	1015 25	1157 13	.0085
8 14	4	153 07	1007 22	1160 62	0112
10 18	5	162.23	1000.72	1163.44	0137
12 22	6	170 12	995.24	1165.82	.0163
14 25	7	176.91	990 47	1167.89	0189
16.29	8	182.91	986.24	1169.72	.0214
18 32	9	188 31	982.43	1171 37	.0239
20.36	10	193 24	978.95	1172 87	.0264
22.39	11	197 76	975 76	1174.25	.0289
24 43	12	201.96	972.80	1175.53	.0313
26 46	13	205 88	970 02	1176 73	.0337
28 51	14	209 56	967.42	1177.85	0362
Gauge Press.	14.7	212 00	965 7	1178.10	.0380
3	15	213.02	964 97	1178.91	0387
1 3	16	216.29	962.65	1179.90	0413
2 3	17	219 41	960 45	1180 85	.0437
3.3	18	222.37	958 34	1181 76	.0462
4.3	19	225.20	956 34	1182.61	.0487
5.3	20	227.91	954 41	1183 45	.0511
6.3	21	230 51	952 57	1184.24	.0536
7 3	22	233.01	950.79	1185.00	.0561
8.3	23	235.43	949 07	1185.74	.0585
9.3	24	237.75	947 42	1186 45	.0610
10.3	25	240.00	945 82	1187.13	.0634
11 3	26	242.17	944.27	1187 80	.0658
12 3	27	244 28	942 77	1188.44	0683
13.3	28	246 32	941 32	1189.06	.0707
14.3	29	248.31	939 90	1189 67	0731
15.3	30	250 24	938.92	1190 26	.0755
16 3	31	252 12	937.18	1190.83	.0779
17.3	32	253.95	935 88	1191 39	0803
18.3	33	255.73	934.60	1191.93	0827
19 3	34	257.47	933.36	1192.46	.0851
20.3	35	259.17	932 15	1192 98	.0875
21.3	36	260.83	930 96	1193.49	.0899
22.3	37	262.	929 80	1193.98	.0922
23.3	38	264.04	928.67	1194 47	.0946
24 3	39	265.59	927.56	1194.94	0970
25 3	40	267.12	926.47	1195.41	.0994
26.3	41	268.61	925.40	1195.86	.1017
27 3	42	270.07	924 35	1196 31	.1041

TABLE NO. 24.

Properties of Saturated Steam — Continued.

Gauge Pressure.	Pressure Above Zero.	Temperature.	Latent Heat.	Total Heat. Above Zero.	Weight of one Cubic Foot.
Pounds per Square Inch.	Pounds per Square Inch.	Fahr. Deg.	B. T. U.	Fahr., B. T. U.	Pounds.
28.3	43	271.50	923.33	1196.74	.1064
29.3	44	272.91	922.32	1197.17	.1088
30.3	45	274.29	921.33	1197.60	.1111
31.3	46	275.65	920.36	1198.01	.1134
32.3	47	276.98	919.40	1198.42	.1158
33.3	48	278.29	918.46	1198.82	.1181
34.3	49	279.58	917.54	1199.21	.1204
35.3	50	280.85	916.63	1199.60	.1227
36.3	51	282.09	915.73	1199.98	.1251
37.3	52	283.32	914.85	1200.35	.1274
38.3	53	284.53	913.98	1200.72	.1297
39.3	54	285.72	913.13	1201.08	.1320
40.3	55	286.89	912.29	1201.44	.1343
41.3	56	288.05	911.46	1201.79	.1366
42.3	57	289.11	910.64	1202.14	.1388
43.3	58	290.31	909.83	1202.48	.1411
44.3	59	291.42	909.03	1202.82	.1434
45.3	60	292.52	908.24	1203.15	.1457
46.3	61	293.59	907.47	1203.48	.1479
47.3	62	294.66	906.70	1203.81	.1502
48.3	63	295.71	905.94	1204.13	.1524
49.3	64	296.75	905.20	1204.44	.1547
50.3	65	297.77	904.46	1204.76	.1569
51.3	66	298.78	903.73	1205.07	.1592
52.3	67	299.78	903.01	1205.37	.1614
53.3	68	300.77	902.29	1205.67	.1637
54.3	69	301.75	901.59	1205.97	.1659
55.3	70	302.71	900.89	1206.26	.1681
56.3	71	303.67	900.21	1206.56	.1703
57.3	72	304.61	899.52	1206.84	.1725
58.3	73	305.55	898.85	1207.13	.1748
59.3	74	306.47	898.18	1207.41	.1770
60.3	75	307.38	897.52	1207.69	.1792
61.3	76	308.29	896.87	1207.96	.1814
62.3	77	309.18	896.23	1208.24	.1836
63.3	78	310.06	895.59	1208.51	.1857
64.3	79	310.94	894.95	1208.77	.1879
65.3	80	311.81	894.33	1209.04	.1901
66.3	81	312.67	893.70	1209.30	.1923
67.3	82	313.52	893.09	1209.56	.1945
68.3	83	314.36	892.48	1209.82	.1967
69.3	84	315.19	891.88	1210.07	.1988
70.3	85	316.02	891.28	1210.32	.2010
71.3	86	316.83	890.69	1210.57	.2032
72.3	87	317.65	890.10	1210.82	.2053

TABLE NO. 25.

Properties of Saturated Steam — Continued.

Gauge Pressure.	Pressure Above Zero.	Temperature.	Latent Heat.	Total Heat. Above Zero.	Weight of One Cubic Foot.
Pounds per Square Inch.	Pounds per Square Inch.	Fahr. Deg.	B. T. U.	Fahr., B. T. U.	Pounds.
73.8	88	318.45	889.52	1211.06	.2075
74.8	89	319.24	888.94	1211.31	.2097
75.3	90	320.03	888.37	1211.55	.2118
76.3	91	320.82	887.80	1211.79	.2130
77.3	92	321.59	887.24	1212.02	.2160
78.3	93	322.36	886.68	1212.26	.2182
79.3	94	323.12	886.13	1212.49	.2204
80.3	95	323.88	885.58	1212.72	.2224
81.3	96	324.63	885.04	1212.95	.2245
82.3	97	325.37	884.50	1213.18	.2266
83.3	98	326.11	883.97	1213.40	.2288
84.3	99	326.84	883.44	1213.62	.2309
85.3	100	327.57	882.91	1213.84	.2330
86.3	101	328.29	882.39	1214.06	.2351
87.3	102	329.00	881.87	1214.28	.2371
88.3	103	329.71	881.35	1214.50	.2392
89.3	104	330.41	880.84	1214.71	.2413
90.3	105	331.11	880.34	1214.92	.2434
91.3	106	331.80	879.84	1215.14	.2454
92.3	107	332.49	879.34	1215.35	.2475
93.3	108	333.17	878.84	1215.55	.2496
94.3	109	333.85	878.35	1215.76	.2516
95.3	110	334.52	877.86	1215.97	.2537
96.3	111	335.19	877.37	1216.17	.2558
97.3	112	335.85	876.89	1216.37	.2578
98.3	113	336.51	876.41	1216.57	.2599
99.3	114	337.16	875.94	1216.77	.2619
100.3	115	337.81	875.47	1216.97	.2640
101.3	116	338.45	875.40	1217.17	.2661
102.3	117	339.10	874.53	1217.36	.2681
103.3	118	339.73	874.07	1217.56	.2702
104.3	119	340.36	873.61	1217.75	.2722
105.3	120	340.99	873.15	1217.94	.2742
106.3	121	341.61	872.70	1218.13	.2762
107.3	122	342.23	872.25	1218.32	.2782
108.3	123	342.85	871.80	1218.51	.2802
109.3	124	343.46	871.35	1218.69	.2822
110.3	125	344.07	870.91	1218.88	.2842
111.3	126	344.67	870.47	1219.06	.2862
112.3	127	345.27	870.03	1219.27	.2882
113.3	128	345.87	869.59	1219.43	.2902
114.3	129	346.45	869.16	1219.61	.2922
115.3	130	347.05	868.73	1219.79	.2942
116.3	131	347.64	868.30	1219.97	.2961

TABLE NO. 26.

Properties of Saturated Steam — Continued.

Gauge Pressure.	Pressure Above Zero.	Temperature.	Latent Heat.	Total Heat. Above Zero.	Weight of One Cubic Foot.
Pounds per Square Inch.	Pounds per Square Inch.	Fahr. Deg.	B. T. U.	Fahr., B. T. U.	Pounds.
117.3	132	348.22	867.88	1220.15	.2981
118.3	133	348.80	867.46	1220.32	.3001
119.3	134	349.38	867.03	1220.50	.3020
120.3	135	349.95	866.62	1220.67	.3040
121.3	136	350.52	866.20	1220.85	.3060
122.3	137	351.08	866.79	1221.02	.3079
123.3	138	351.75	865.38	1221.19	.3099
124.3	139	352.21	864.97	1221.36	.3118
125.3	140	352.76	864.56	1221.53	.3138
126.3	141	353.31	864.16	1221.70	.3158
127.3	142	353.86	863.76	1221.87	.3178
128.3	143	354.41	863.36	1222.03	.3199
129.3	144	354.96	862.96	1222.20	.3219
130.3	145	355.50	862.56	1222.36	.3239
131.3	146	356.03	862.17	1222.53	.3259
132.3	147	356.57	861.78	1222.69	.3279
133.3	148	357.10	861.39	1222.85	.3299
134.3	149	357.63	861.00	1223.01	.3319
135.3	150	358.16	860.62	1223.18	.3340
136.3	151	358.68	860.23	1223.33	.3358
137.3	152	359.20	859.85	1223.49	.3376
138.3	153	359.72	859.47	1223.65	.3394
139.3	154	360.23	859.10	1223.81	.3412
140.3	155	360.74	858.72	1223.97	.3430
141.3	156	361.26	858.35	1224.12	.3448
142.3	157	361.76	857.98	1224.28	.3466
143.3	158	362.27	857.61	1224.43	.3484
144.3	159	362.77	857.24	1224.58	.3502
145.3	160	363.27	856.87	1224.74	.3520
146.3	161	363.77	856.50	1224.89	.3539
147.3	162	364.27	856.14	1225.04	.3558
148.3	163	364.76	855.78	1225.19	.3577
149.3	164	365.25	855.42	1225.34	.3596
150.3	165	365.74	855.06	1225.49	.3614
151.3	166	366.23	854.70	1225.64	.3633
152.3	167	366.71	854.35	1225.78	.3652
153.3	168	367.19	853.99	1225.93	.3671
154.3	169	367.68	853.64	1226.08	.3690
155.3	170	368.15	853.29	1226.22	.3709
156.3	171	368.63	852.94	1226.37	.3727
157.3	172	369.10	852.59	1226.51	.3745
158.3	173	369.57	852.25	1226.66	.3763
159.3	174	370.04	851.90	1226.80	.3781
160.3	175	370.51	851.56	1226.94	.3799
161.3	176	370.97	851.22	1227.08	.3817

TABLE NO. 27.

Properties of Saturated Steam — Continued.

Gauge Pressure.	Pressure Above Zero.	Temperature.	Latent Heat.	Total Heat. Above Zero.	Weight of One Cubic Foot.
Pounds per Square Inch.	Pounds per Square Inch.	Fahr. Deg.	B. T. U.	Fahr., B. T. U.	Pounds.
162.3	177	371.44	850.88	1227.32	.3835
163.3	178	371.90	850.54	1227.37	.3853
164.3	179	372.36	850.20	1227.51	.3871
165.3	180	372.82	849.86	1227.65	.3889
166.3	181	373.27	849.53	1227.78	.3907
167.3	182	373.73	849.20	1227.92	.3925
168.3	183	374.18	848.86	1228.06	.3944
169.3	184	374.63	848.53	1228.20	.3962
170.3	185	375.08	848.20	1228.33	.3980
171.3	186	375.52	847.88	1228.47	.3999
172.3	187	375.97	847.55	1228.61	.4017
173.3	188	376.41	847.22	1228.74	.4035
174.3	189	376.85	846.90	1228.87	.4053
175.3	190	377.29	846.58	1229.01	.4072
176.3	191	377.72	846.23	1229.14	.4089
177.3	192	378.16	845.94	1229.27	.4107
178.3	193	378.59	845.62	1229.41	.4125
179.3	194	379.02	845.30	1229.54	.4143
180.3	195	379.45	844.99	1229.67	.4160
181.3	196	379.97	844.68	1229.80	.4178
182.3	197	380.30	844.36	1229.93	.4196
183.3	198	380.72	844.05	1230.06	.4214
184.3	199	381.15	843.74	1230.19	.4231
185.3	200	381.57	843.43	1230.31	.4249
186.3	201	381.99	843.12	1230.44	.4266
187.3	202	382.41	842.81	1230.57	.4283
188.3	203	382.82	842.50	1230.70	.4300
189.3	204	383.24	842.20	1230.82	.4318
190.3	205	383.65	841.89	1230.95	.4335
191.3	206	384.06	841.59	1231.07	.4352
192.3	207	384.47	841.29	1231.20	.4369
193.3	208	384.88	840.99	1231.32	.4386
194.3	209	385.28	840.69	1231.45	.4403
195.3	210	385.67	840.39	1231.57	.4421

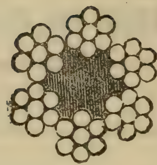


TABLE NO. 28.

SWEDES IRON TRANSMISSION OR HAULAGE ROPES.

7 Wires to the Strand. Hemp Centre.

Diameter in Inches.	Price per Foot in Cents.	Breaking Strain in Tons of 2,000 Pounds.	Proper Working Load in Tons of 2,000 Lbs.		Average Weight per Foot.	Minimum Size of Drums or Sheaves in Ft.
9-32	3¼	1.4	¼	These Ropes are principally used for Derricks, Guys, Steamboat Rigging, Ferries, Transmission of Power, etc.	.12½	1½
5-16	3¾	1.7	⅜		.15	1¾
⅜	4½	2 4	½		.22	2
7-16	5⅛	3 3	⅝		.30	2¼
½	6⅛	4 2	⅞		.39	2½
9-16	8	5.3	1		.50	3
⅝	10	6.6	1⅜		.64	3½
11-16	12	7.9	1½		.75	4
¾	14	9.3	1⅞		.89	4½
⅞	17½	12.	2 2-5		1.20	5
1	23	16.	3 1-5		1.58	5¾
1⅛	29	20	4		2 00	6¼
1¼	36	24.	4 4-5		2 45	7¼
1⅜	43	29.	5 4-5		3 00	8
1½	51	34.	7		3 55	8½

CRUCIBLE CAST STEEL TRANSMISSION OR HAULAGE ROPES.

7 Wires to the Strand. Hemp Centre.

Diameter in Inches.	Price per Foot in Cents.	Breaking Strain in Tons of 2,000 Pounds.	Proper Working Load in Tons of 2,000 Pounds.		Average Weight per Foot.	Minimum size of Drums or Sheaves in Ft.
9-32	4.	2¾	½	These Ropes are principally used for Derricks, Guys, Steamboat Rigging, Ferries, Transmission of Power, etc., etc.	12½	2¼
5-16	4½	3½	¾		.15	2½
⅜	5½	5	1		.22	2¾
7 16	6½	6½	1¼		.30	3¼
½	7½	8½	1¾		.39	4
9-16	9	10½	2		.50	4½
⅝	11	13	2½		.64	5¼
11-16	13½	16	3		.75	6
¾	16	19	3¾		.89	6¾
⅞	22	24	5		1 20	7½
1	28	32	6½		1 58	8½
1⅛	36	40	8		2 00	9½
1¼	43	48	10		2 45	10¾
1⅜	51	58	12		3 00	12
1½	60	68	14		3 55	13

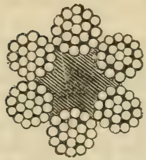


TABLE NO. 29.

CRUCIBLE CAST STEEL PL LE HOISTING ROPES.

19 Wires to the Strand. Hemp Centre.

Diameter in Inches.	Price per Foot in Cents.	Breaking Strain in Tons of 2,000 Pounds.	Diam. of Hemp Rope of Equal Strength.	Proper Working Load in Tons of 2,000 Lbs.	Average Weight per Foot.	Minimum Size of Drums or Sheaves in Ft.
1/4	9	2 1/2	3/4	1/2	.10	3/4
5-16	9 1/4	3 1/2	1	3/4	.15	1
3/8	9 1/2	5	1 1/8	1	.22	1 1/2
7-16	10	7	1 1/4	1 2-5	.30	2
1/2	11	9	1 1/2	1 4-5	.39	2 1/4
9-16	12	11	1 9-16	2 1-5	.50	2 3/4
5/8	14	14	1 3/4	2 4-5	.64	3 1/2
3/4	18	20	2 1/8	4	.89	4
7/8	23	26	2 5/8	5 1-5	1.20	4 1/2
1	30	34	3	6 4-5	1 58	5 1/4
1 1/8	38	42	3 3/8	8 2-5	2.00	6
1 1/4	46	52	3 3/4	10	2.45	6 1/2
1 3/8	56	63	4	12	3.00	7
1 1/2	66	77	4 1/4	15	3.55	7 1/2
1 5/8	74	86	4 3/4	17	4.15	8 1/2
1 3/4	93	106	5 1/4	20	5.25	10
2	111	125	.....	25	6.30	12
2 1/4	142	160	.....	32	8 00	13

SWEDES IRON PLIABLE HOISTING ROPES.

ires to the Strand. Hemp Centre.

Diameter in Inches.	Price per Foot in Cents.	Breaking Strain in Tons of 2,000 Pounds.	Diam. of Hemp Rope of Equal Strength.	Proper Working Load in Tons of 2,000 Lbs.	Average Weight per Foot.	Minimum Size of Drums or Sheaves in Ft.
1/4	6 1/2	1.20	9-16	1/4	.10	1/2
5-16	6 3/4	1.70	5/8	3/8	15	2/3
3/8	7	2.50	7/8	1/2	.22	1
7-16	7 1/2	3.40	1	5/8	.30	1 1/2
1/2	8	4.40	1 1/8	7/8	.39	1 3/4
9-16	10	5.50	1 5-16	1 1-10	.50	2
5/8	12	6 80	1 1/2	1 3/8	.64	2 1/2
3/4	16	9.70	1 5/8	1 7/8	.89	3
7/8	20	13.	2	2 3-5	1 20	3 1/2
1	26	17.	2 1/4	3 2-5	1.58	4
1 1/8	33	21.	2 5/8	4 1-5	2 00	4 1/2
1 1/4	40	25.	3 1/8	5	2.45	5
1 3/8	48	31.	3 1/4	6 1-5	3.00	6
1 1/2	57	36.	3 1/2	7 1-5	3.55	6 1/2
1 5/8	63	42.	4	8 2-5	4 15	7
1 3/4	80	48.	4 1/4	9 3-5	5.25	7 1/2
2	92	62.	4 3/4	12 2-5	6.30	9
2 1/4	117	78.	5 1/3	15 3-5	8 00	10

TABLE NO. 30.

19 Wires to the Strand. Hemp Centre.

Diameter in Inches.	Price per Foot, in Cents.	Breaking Strain in Tons of 2,000 Pounds.	Proper Working Load in Tons of 2,000 Lbs.	Average Weight per Foot.	Minimum Size of Drums or Sheaves in Ft.
3/8	14	8	1 1/4	.22	2
7-16	15	10	1 3/4	.30	2 1/2
1/2	17	13	2	.39	2 3/4
9-16	19	15	2 1/2	.50	3
5/8	22	20	3 1/4	.64	4
3/4	28	30	5	.89	5
7/8	35	40	6 1/2	1.20	6
1	45	50	8 1/2	1.58	7
1 1/8	54	63	11	2.00	8
1 1/4	65	76	13	2 45	9
1 3/8	80	95	16	3.00	10
1 1/2	95	115	19	3.55	11
1 5/8	112	130	22	4.15	12
1 3/4	136	160	25	5.25	13
2	160	220	33	6 30	14
2 1/4	220	235	40	8.00	15

7 Wires to the Strand. Hemp Centre.

1/2	12 1/2	11	2	.39	3
9-16	15	14	2 1/4	.50	4
5/8	18	18	3	.64	5 1/4
3/4	25	28	5	.89	6
7/8	33	36	7	1.20	7
1	42	48	9 1/2	1.58	8
1 1/8	53	60	11	2.00	9
1 1/4	64	72	13	2 45	10
1 3/8	76	82	16	3.00	11
1 1/2	92	100	20	3 55	12

WIRE ROPE FOR INCLINED PLANES.

For the benefit of those desiring to use wire rope on slopes, inclined Planes, etc , a table by which the strain produced by any load may be readily ascertained.

This table gives only the strain produced on a rope by a load of one ton of two thousand pounds, an allowance for rolling friction being made. An additional allowance for the weight of the rope will have to be made.

Example: For an inclination of 100 feet in 100 feet, corresponding to an angle of 45°, a load of 2,000 pounds will produce a strain on the rope of 1,419 pounds, and for a load of 9,000 pounds the strain on the rope will be $\frac{1419 \times 9000}{2000} = 6{,}385\frac{1}{2}$ pounds.

Elevation in 100 Feet.	Corresponding Angle of Inclination in Degrees.	Strain in lbs. on Rope from a load of 2,000 lbs.	Elevation in 100 Feet.	Corresponding Angle of Inclination in Degrees.	Strain in lbs. on Rope from a load of 2,000 lbs.
5	2 7/8	112	95	43 1/2	1,385
10	5 1/3	211	100	45	1,419
15	8 1/3	308	105	46 1/2	1,457
20	11 1-5	404	110	47 3/4	1,487
25	14 1/2	497	115	49	1,516
30	16 3/4	586	120	50 1/4	1,544
35	19 1-5	673	125	51 1/2	1,570
40	21 5-6	754	130	52 1/2	1,592
45	24 1/4	832	135	53 1/2	1,614
50	26 1/3	905	140	54 1/2	1,633
55	28 5-6	975	145	55 1/2	1,653
60	31	1,040	150	56 1/4	1,671
65	33 1-12	1,100	155	57 1/4	1,689
70	35	1,156	160	58	1,703
75	37	1,210	165	58 4-5	1,717
80	38 2/3	1,260	170	59 1/2	1,729
85	40 1/2	1,304	175	60 1/4	1,742
90	42	1,347			

A factor of safety of five to seven times should be taken; that is, the working load on the rope should only be one-fifth to one-seventh of its breaking strength. As a rule, ropes for shafts should have a factor of safety of five, and on inclined planes, where the wear is much greater, the factor of safety should be seven.

TABLE NO. 31.

Table of Transmission of Power by Wire Ropes.

This table is based upon scientific calculations, careful observations and experience, and can be relied upon when the distance exceeds 100 feet. We also find by experience that it is best to run the wire rope transmission at *the medium number of revolutions* indicated in the table, as it makes the best and smoothest running transmission. If more power is needed than is indicated at 80 to 100 revolutions, choose a larger diameter of sheave.

Diameter of Sheave in ft.	Number of Revolutions.	Diameter of Rope.	Horse-Power.	Diameter of Sheave in Ft.	Number of Revolutions.	Diameter of Rope.	Horse-Power.
3	80	$\frac{3}{8}$	3	7	140	9-16	35
3	100	$\frac{3}{8}$	$3\frac{1}{2}$	8	80	$\frac{5}{8}$	26
3	120	$\frac{3}{8}$	4	8	100	$\frac{5}{8}$	32
3	140	$\frac{3}{8}$	$4\frac{1}{2}$	8	120	$\frac{5}{8}$	39
4	80	$\frac{3}{8}$	4	8	140	$\frac{5}{8}$	45
4	100	$\frac{3}{8}$	5	9	80	9-16 $\frac{5}{8}$	47 48
4	120	$\frac{3}{8}$	6	9	100	9-16 $\frac{5}{8}$	58 60
4	140	$\frac{3}{8}$	7	9	120	9-16 $\frac{5}{8}$	69 73
5	80	7-16	9	9	140	9 16 $\frac{5}{8}$	82 84
5	100	7-16	11	10	80	$\frac{5}{8}$ 11-16	64 68
5	120	7-16	13	10	100	$\frac{5}{8}$ 11-16	80 85
5	140	7 16	15	10	120	$\frac{5}{8}$ 11-16	96 102
6	80	$\frac{1}{2}$	14	10	140	$\frac{5}{8}$ 11-16	112 119
6	100	$\frac{1}{2}$	17	12	80	11-16 $\frac{3}{4}$	93 99
6	120	$\frac{1}{2}$	20	12	100	11 16 $\frac{3}{4}$	116 124
6	140	$\frac{1}{2}$	23	12	120	11-16 $\frac{3}{4}$	140 149
7	80	9-16	20	12	140	$\frac{7}{8}$	173
7	100	9-16	25	14	80	1 $1\frac{1}{8}$	141 148
7	120	9-16	30	14	100	1 $1\frac{1}{8}$	176 185

TABLE NO. 32.

HORIZONTAL STATIONARY SLIDE VALVE ENGINES.

Table Showing the Horse-Power Developed at Different Speeds and Pressures of Steam.

Size of Cylinder.	7 x 10			8 x 10			9 x 12			10 x 12			10 x 14			11 x 14			12 x 16			13 x 16		
Revolutions.	180	200	220	180	200	220	180	200	220	180	200	220	160	180	220	160	180	200	140	160	180	140	160	180
Horse-power with 80 lb. steam........	10	12	13	14	15	17	19	21	23	23	26	28	26	28	31	30	34	37	36	40	44	44	49	53
Horse-power with 90 lb. steam	12	13	14	15	17	18	20	23	26	26	28	31	28	31	34	33	37	40	40	44	49	49	53	60
Horse.power with 100 lb. steam	13	14	15	17	18	20	23	25	28	28	31	34	31	34	38	36	40	44	44	49	54	53	60	64

In the above table the fractional parts of horse-power have been dropped, the nearest whole number being sufficiently accurate to illustrate the fact that increased boilure pressure, or increased piston speed, gives increase power.

Horse Power.	Size of Cylinder. Diameter, Inches.	Size of Cylinder. Stroke, Inches.	Revolutions per Minute.	Horse Power.	Size of Cylinder. Diameter Inches.	Size of Cylinder. Stroke Inches.	Revolutions per Minute.	Bore of Cylinder.	Stroke.	Revolutions per Minute.	Piston Speed per minute	Indicated H. P. at 40 lb. Average Piston Pressure.
4	4	5	250	15	8	8	200	10	16	188	500 ft.	47
5	5	5	225	25	10	10	180	12	16	188	"	68
7	6	6	225	35	12	10	180	13	18	168	"	80
10	7	7	200	45	12	12	180	14	18	167	"	93
15	8	8	200	60	14	12	180	14	20	150	"	93
20	9	9	80	85	15	16	160	16	20	150	"	121
				110	17	16	160	18	24	125	"	154
								20	24	125	"	190
								22	30	100	"	230
								24	30	100	"	274
								24	36	84	"	274
								26	36	84	"	321
								30	42	72	"	425

TABLE NO. 33.

Percentage of Power Gained by Adding a Condenser, the Speed and Point of Cut-Off Remaining the Same.*

VACUUM = 24.5 INCHES.

M. E. P.	Per Cent of Power Gained.	M. E. P.	Per Cent of Power Gained.	M. E. P.	Per Cent of Power Gained.	M. E. P.	Per Cent of Power Gained.
5	250	17	70.5	38	31.5	62	19.3
6	200	18	66.6	40	30	64	18.7
7	171.4	19	63.1	42	28.5	66	18.1
8	150	20	60	44	27.2	68	17.6
9	133.3	22	54.5	46	26.08	70	17.1
10	120	24	50	48	25	72	16.6
11	109	26	46.1	50	24	74	16 2
12	100	28	44.3	52	23.07	76	15.7
13	92.3	30	40	54	22.2	78	15 3
14	85.7	32	37.5	56	21 4	80	15
15	80	34	35.2	58	20.6	82	14.63
16	75	36	33.3	60	20	85	14.12

* Calculated gain due to vacuum, friction not considered.

TABLE NO. 34.

Sizes of Cylinders Usually Furnished for Compound Pumps, and Corresponding Ratios of Expansion.

Diameter High Pressure Cylinder.	Diameter Low Pressure Cylinder.	Ratio of Expansion.
6	10	2.78
7	12	2.94
8	12	2.25
9	14	2.42
10	16	2.56
12	18	2.25
14	20	2.04
16	24	2.25
18	30	2.77

TABLE NO. 35.

Indicated Horse-power for One Pound of Mean Pressure.

Diameter of Cylinder in Inches.	Speed of Piston in Feet per Minute.							
	240	300	350	400	450	500	550	600
6	0·205	0·256	0·299	0·342	0·385	0·428	0·471	0·513
7	0·279	0·348	0·408	0·466	0·524	0·583	0 641	0·699
8	0·365	0·456	0·532	0·608	0·685	0·761	0·837	0·912
9	0·462	0·577	0·674	0·770	0 866	0·963	1·059	1·154
10	0·571	0·714	0·833	0·952	1·071	1·190	1·309	1·428
11	0·691	0·864	1·008	1·153	1·296	1·44	1·584	1·728
12	0·820	1·025	1·195	1·366	1·540	1·708	1·880	2·050
13	0·964	1·206	1·407	1·608	1·809	2·01	1·211	2·412
14	1·119	1·398	1·631	1·864	2·097	2·331	2·564	2·797
15	1·285	1·606	1·873	2·131	2·409	2·677	2·945	3·212
16	1·461	1·827	2·131	2·436	2·741	3·045	3·349	3·654
17	1·643	2·054	2·396	2·739	3·081	3·424	3·766	4·108
18	1·849	2·312	2·697	3·083	3·468	3·854	4·239	4·624
19	2·061	2·577	3·006	3·436	3·865	4·297	4·724	5·154
20	2·292	2·855	3·331	3·807	4·285	4·759	5·234	5·731
21	2·518	3·148	3·672	4·197	4·722	5·247	5·771	6·296
22	2·764	3·455	4·031	4·607	5·183	5·759	6·334	6·911
23	3·021	3·776	4·405	5·035	5·664	6·294	6·923	7·552
24	3·289	4·111	4·797	5·482	6·167	6·853	7·538	8·223
25	3·569	4·461	5·105	5·948	6·692	7·436	8·179	8·923
26	3·861	4·826	5·630	6·435	7·239	8·044	8·848	9·652
27	4·159	5·199	6·066	6·932	7·799	8·666	9·532	10·399
28	4·477	5·596	6·529	7·462	8·395	9·328	10 261	11·193
29	4·805	6·006	7·007	8·008	9·009	10·01	11·011	12·012
30	5·141	6·426	7·497	8·568	9·639	10·71	11·781	12·852
31	5·486	6·865	8·001	9·144	10·287	11·43	12·573	13·716
32	5·846	7·308	8·526	9·744	10·962	12·18	13·398	14·616
33	6·216	7·770	9·065	10·360	11·655	12·959	14·245	15·54
34	6·59	8·238	9·611	10·984	12·357	13·73	15·103	16·476
35	6·993	8·742	10·199	11·656	13·113	14·57	16·027	17·484
36	7·401	9·252	10·794	12·336	13·878	15·42	16·962	18·504
37	7·819	9·774	11·403	13·032	14·861	16·29	17·519	19·548
38	8·246	10·308	12·026	13·744	15·462	17·18	18·898	20·616
39	8·618	10·86	12·67	14·48	16·29	18·1	19·91	21·61
40	9·139	11·424	13·328	15·232	17·136	19·04	20·944	22·848
41	9·604	12·006	14·007	16·008	18·009	20·00	22·011	24·012
42	10·065	12·594	14·693	16·792	18·901	20·99	23·089	25·188
43	10·56	13·20	15·4	17·6	19·8	22·00	24 2	26·4
44	11·046	13·818	16·121	18·424	20·727	23·03	25·333	27·636
45	11·563	14·454	16·863	19·272	21·681	24·09	26·399	28·908
46	12·086	15·128	17·626	20·144	22·662	25·18	27·698	30·216
47	12·614	15·768	18·396	21 024	23·652	26·28	28·908	31·536
48	12·816	16·446	19·187	21·928	24·669	27·41	30·151	32·152
49	12·913	17·142	19·999	22·856	25·713	28·57	33·427	34·284
50	14·28	17·85	20·825	23·8	26·775	29·75	32·725	35·7

NOTE. — **Mean effective pressure = Mean pressure minus the back pressure.**

TABLE NO. 36.

Table of Mean Absolute Pressures.

Absolute Initial Pressure in lb. per Square Inch.	CUT-OFF IN FRACTIONS OF THE STROKE.															
	1/10	1/8	1/6	1/5	1/4	3/10	1/3	4/10	1/2	6/10	2/3	7/10	3/4	8/10	7/8	9/16
	RATIO OF EXPANSION.															
	10	8	6	5	4	3.33	3	2.5	2	1.66	1.5	1 43	1.33	1.25	1.14	1.11
10	3·303	3·848	4·648	5·218	5·965	6·609	6·989	7·664	8·465	9·037	9·366	9·489	9·654	9·784	9·921	9·946
15	4·954	5·774	6·980	7·827	8·947	9·913	10·49	11·49	12·69	13·55	14·05	14·23	14·48	14·67	14·88	14·92
20	6·606	7·697	9·297	10·43	11·93	13·21	13·98	15·32	16·93	18·07	18·73	18·98	19·30	19·56	19·84	19·89
25	8·255	9·620	11·63	13·04	14·91	16·52	17·48	19·16	21·16	22·59	23·41	23·72	24·13	24·46	24·80	24·86
30	9·909	11·54	13·96	15·65	17·90	19·82	20·99	22·99	25·39	27·11	28·10	28·46	28·96	29·35	29·76	29·84
35	11·56	13·47	16·28	18·26	20·87	23·13	24·48	26·82	29·64	31·62	32·78	33·21	33·79	34·24	34·72	34·81
40	13·21	15·39	18·59	20·87	23·86	26·43	27·98	30·65	33·86	36·15	37·43	37·95	38·61	39·13	39·68	39·78
45	14·86	17·32	20·94	23·48	26·84	29·74	31·48	34·48	38·09	40·66	42·14	42·70	43·44	44·02	44·64	44·75
50	16·51	19·24	23·26	26·09	29·82	33·04	34·96	38·32	42·32	45·18	46·82	47·44	48·27	48·92	49·60	49·73
60	19·81	23·09	27·92	31·31	35·79	39·65	41·98	45·98	50·79	54·22	56·20	56·93	57·92	58·70	59·52	59·67
70	23·10	26·94	32·55	36·53	41·75	46·26	48·96	53·65	59·25	63·26	65·56	66·42	67·57	68·48	69·44	69·62
80	26·42	30·79	37·21	41·74	47·72	52·87	55·96	61·31	67·72	72·30	74·86	75·91	77·23	78·27	79·37	79·56
90	29·73	34·64	41·88	46·96	53·68	59·48	62·97	68·97	76·18	81·34	84·28	85·40	86·88	88·05	89·28	89·50
100	33·03	38·48	46·51	52·18	59·65	66·08	69·92	76·64	84·65	90·36	93·64	94·89	96·54	97·84	99·21	99·46
110	36·33	42·34	51·17	57·40	65·61	72·70	76·94	84·30	93·11	99·40	103·0	104·4	106·2	107·6	109·1	109·4
120	39·64	46·18	55·84	62·62	71·58	79·31	83·96	91·96	101·6	108·4	112·4	113·8	115·8	117·4	119·0	119·3
130	42·94	50·03	60·47	67·83	77·54	85·91	90·92	99·63	110·0	117·5	121·7	123·3	125·5	127·2	128·9	129·3
140	46·23	53·88	65·13	73·05	83·51	92·52	97·94	107·3	118·5	126·5	131·1	132·8	135·1	136·9	138·8	139·2
150	49·54	57·73	69·80	78·27	89·47	99·13	104·9	114·9	126·9	135·5	140·5	142·3	144·8	146·7	148·8	149·2
200	66·06	76·97	93·05	104·4	119·3	132·2	139·9	153·3	169·3	180·7	187·3	189·8	193·0	195·6	198·4	198·9

NOTE.—To find the mean *effective* pressure, deduct the mean absolute *back* pressure from the pressure given in the table.

TABLE NO. 37.

SIZES AND DIMENSIONS OF STANDARD CORLISS ENGINES.

Size of Cylinder		No. of Revolutions.	Indicated Horse Power.									Fly Wheel.			Crank Shaft					Steam Pipes.	
			80 lbs. Pressure.			90 lbs. Pressure.			100 lbs. Pressure												
Diameter.	Stroke.		1-5 Cut-off.	1-4 Cut-off.	1-3 Cut-off.	1-5 Cut-off.	1-4 Cut-off.	1-3 Cut-off.	1-5 Cut-off.	1-4 Cut-off.	1-3 Cut-off.	Diameter.	Face.	Weight in Pounds.	Diameter.	Length.	Center Shaft above Found.	Center Shaft to Back of Cylinder Head.		Diameter Steam.	Diameter Exhaust.
In.	In.											Ft.	In.		In.	Ft.	In.	Ft.	In.	In.	In.
10	24	90	30	36	43	34	41	48	39	49	60	8	13	3,500	5	6	25	10	11	3	4
10	30	90	37	45	53	42	51	60	48	58	68	8	13	4,300	5	6	25	13		3	4
12	30	90	54	65	76	62	74	87	69	83	97	9	15	5,700	6	6½	25	13	4	3½	4½
12	36	85	61	73	86	70	84	98	78	94	110	10	17	6,300	6	7	25	15	5	3½	4½
14	36	85	83	100	118	95	114	134	107	128	150	10	19	8,900	7	7½	25½	15	6	4	5
14	42	82	93	112	132	107	128	140	120	144	168	12	19	8,900	7	7½	25½	17	7	4	6
16	36	82	105	126	140	120	144	167	135	162	187	12	21	10,500	8	8	27	15	9	4½	6
16	42	78	116	139	163	133	159	188	150	179	210	12	23	11,800	8	8	27	17	10	4½	6
18	36	80	129	155	182	148	177	208	166	199	233	12	25	12,600	9	9	30	15	11	5	7
18	42	78	147	176	208	168	202	237	189	227	266	14	25	13,300	9	9	30	18		5	7
18	48	75	162	194	227	185	222	259	208	249	291	15	25	15,300	9	9	30	20	1	5	7
20	42	75	175	210	246	200	240	280	225	270	314	15	29	16,600	10	9	30	18	4	6	8
20	48	72	192	230	268	219	263	307	246	296	346	16	29	18,700	10	9	30	20	5	6	8
22	42	75	211	254	298	242	290	340	271	326	382	16	31	21,000	11	10	32	18	5	6	8
22	48	72	232	278	308	265	318	368	298	358	410	16	31	23,100	11	10	32	20	6	6	8
24	48	70	268	322	380	307	368	432	345	414	486	18	33	24,400	12	11	32	20	11	7	9
24	60	65	311	374	438	356	427	500	401	481	561	18	37	30,500	12	11	32	25	2	7	9
26	48	70	315	378	445	360	432	507	405	486	570	18	37	29,000	13	11½	32	21		7	9
26	60	65	366	439	512	418	502	586	470	564	658	18	37	31,000	13	11½	32	25	3	7	10
28	48	68	355	426	503	406	487	568	457	548	680	18	37	31,000	14	12	34	21	3	8	12
28	54	68	400	477	554	461	546	631	523	616	709	18	37	31,000	14	12	34	23	4	8	12
28	60	65	424	509	594	485	582	655	545	654	763	18	37	31,000	14	12	34	25	6	8	12
30	48	68	407	489	571	466	559	652	524	629	734	22	46	33,000	15	12½	36	21	8	8	12
30	60	62	464	557	650	531	637	743	597	717	837	24	52	38,500	15	12½	36	25	11	8	12
30	72	55	494	593	693	565	678	791	635	762	890	24	60	52,000	15	13	36	30	2		
32	48	65	443	532	622	507	608	710	570	684	800	24	48	34,500	16	13	36	21	11		
32	60	62	529	634	739	604	725	846	680	816	952	24	60	43,900	16	13	36	26	2		
32	72	55	563	675	787	643	772	902	723	868	1013	24	60	58,200	16	14	36	30	5		

TABLE NO. 38.

Speed and Capacity of Centrifugal Pumps.

Diam. Delivery Pipe.	Diam. Suction Pipe.	Capacity in Gals. per Min. 25-ft. Lift.	Horsepower Required for Each Foot Lift.	Side Suction. Revs. per Min.	Double Suction Revs. per Min.
1½	2	75	.05	1450	750
2	2½	120	.07	1150	595
3	3½	250	.14	820	500
4	5	450	.25	725	425
5	6	700	.34	600	380
6	6	1,200	.64	525	345
8	8	2,000	1.10	475	300
10	10	3,000	1.62	450	265
12	12	4,500	2.26	400	255
15	15	7,000	3.50	325	240
18	18	10,000	5.05	285	185
20	20	14,000	7.06	275	140
24	24	18,000	10.20	255	110

The speed for a given capacity varies as the square root of the lift nearly.

TABLE NO. 39.

Diameters of High and Low Pressure Cylinders of Compound Pumps Corresponding to Different Ratios of Piston Areas.

Diam. High Pressure Cylinders.	Ratio of Piston Areas.				
	2 to 1	2½ to 1	3 to 1	3½ to 1	4 to 1
	Diameter of Low Pressure Cylinders.				
6	8½	9½	10½	11	12
7	10	11	12	13	14
8	11	12½	14	15	16
9	13	14	15½	17	18
10	14	16	17	19	20
11	15½	17½	19	20½	22
12	17	19	21	22½	24
13	18½	20½	22½	24½	26
14	20	22	24	26	28
15	21	24	26	28	30
16	22½	25	28	30	32
17	24	27	29½	32	34
18	25½	28½	31	33½	36
19	27	30	33	35½	38
20	28	31½	34½	37½	40
21	30	33	36½	39	42
22	31	35	38	41	44
23	32½	36½	40	43	46
24	34	38	40½	45	48

TABLE NO. 40.
Actual Ratios of Expansion.

Per Cent of Clearance.	Points of Cut-Off. $\frac{1}{10}$	$\frac{1}{8}$	$\frac{1}{5}$	$\frac{1}{4}$	$\frac{3}{10}$	$\frac{1}{3}$	$\frac{3}{8}$	$\frac{4}{10}$	$\frac{5}{10}$	$\frac{6}{10}$	$\frac{5}{8}$	$\frac{7}{10}$	$\frac{3}{4}$	$\frac{8}{10}$	$\frac{7}{8}$	$\frac{9}{10}$
1·0	9·181	7·481	4·809	3·884	3·258	2·944	2·623	2·463	1·983	1·655	1·590	1·422	1·328	1·246	1·141	1·109
1·25	9·0	7·363	4·764	3·875	3·24	2·930	2·612	2·454	1·975	1·653	1·588	1·421	1·327	1·246	1·140	1·109
1·50	8·826	7·25	4·720	3·830	3·222	2·916	2·602	2·445	1·970	1·650	1·585	1·419	1·326	1·245	1·140	1·109
1·75	8·659	7·133	4·677	3·803	3·204	2·902	2·592	2·436	1·966	1·647	1·583	1·418	1·325	1·244	1·140	1·108
2·0	8·5	7·034	4·635	3·777	3·187	2·889	2·582	2·428	1·961	1·645	1·581	1·416	1·325	1·243	1·138	1·108
2·25	8·346	6·932	4·595	3·752	3·170	2·876	2·574	2·420	1·956	1·642	1·579	1·415	1·324	1·243	1·138	1·108
2·50	8·2	6·833	4·555	3·727	3·153	2·863	2·562	2·411	1·952	1·640	1·576	1·413	1·322	1·242	1·138	1·108
2·75	8·088	6·738	4·516	3·702	3·137	2·850	2·552	2·403	1·947	1·637	1·574	1·412	1·321	1·241	1·138	1·107
3·0	7·933	6·645	4·478	3·678	3·121	2·837	2·543	2·395	1·943	1·634	1·572	1·410	1·320	1·240	1·138	1·107
3·25	7·792	6·555	4·440	3·654	3·105	2·824	2·533	2·387	1·938	1·632	1·570	1·409	1·319	1·240	1·138	1·107
3·50	7·666	6·468	4·404	3·631	3·089	2·812	2·524	2·379	1·934	1·629	1·568	1·408	1·318	1·239	1·137	1·106
3·75	7·545	6·390	4·378	3·608	3·074	2·800	2·515	2·371	1·930	1·627	1·566	1·406	1·317	1·238	1·136	1·106
4·0	7·428	6·303	4·333	3·58	3·058	2·788	2·506	2·363	1·925	1·625	1·563	1·405	1·316	1·238	1·136	1·106
4·25	7·315	6·229	4·298	3·564	3·043	2·776	2·497	2·355	1·921	1·622	1·561	1·404	1·315	1·237	1·136	1·106
4·50	7·206	6·147	4·256	3·542	3·028	2·764	2·488	2·348	1·917	1·620	1·559	1·402	1·314	1·236	1·135	1·105
4·75	7·102	6·082	4·232	3·521	3·014	2·752	2·479	2·340	1·913	1·617	1·557	1·401	1·313	1·235	1·135	1·105
5·0	7·0	6·0	4·2	3·5	3·0	2·741	2·470	2·333	1·907	1·615	1·555	1·400	1·312	1·235	1·135	1·105
5·25	6·901	5·985	4·168	3·478	2·986	2·730	2·461	2·325	1·904	1·613	1·553	1·398	1·311	1·234	1·134	1·104
5·50	6·806	5·861	4·130	3·459	2·971	2·719	2·453	2·318	1·900	1·610	1·551	1·397	1·310	1·233	1·134	1·104
5·75	6·714	5·794	4·106	3·439	2·957	2·708	2·445	2·311	1·896	1·608	1·549	1·396	1·309	1·233	1·134	1·104
6·0	6·625	5·729	4·076	3·418	2·944	2·697	2·436	2·304	1·892	1·606	1·547	1·394	1·308	1·232	1·133	1·104
6·25	6·538	5·666	4·047	3·407	2·931	2·686	2·428	2·297	1·888	1·603	1·545	1·393	1·307	1·231	1·133	1·103
6·50	6·454	5·6[illegible]5	4·045	3·380	2·917	2·675	2·420	2·290	1·884	·601	1·543	1·392	1·306	1·231	1·132	1·103
6·75	6·373	5·5[illegible]	3·990	3·362	2·904	2·665	2·412	2·283	1·881	1·599	1·541	1·390	1·305	1·230	1·132	1·103
7·0	6·254	5	3·963	3·342	2·892	2·655	2·404	2·276	1·877	1·597	1·539	1·389	1·304	1·229	1·132	1·103

TABLE NO. 41.

TABLE OF WAGES.

DAYS	$15	$20	$25	$30	$35	$40	$45	$50	$55	$60	$65	$70	$75	$80	$85	$90	$95	$100
1	50	67	83	$1 00	$1 17	$1 33	$1 50	$1 67	$1 83	$2 00	$2 17	$2 33	$2 50	$2 67	$2 83	$3 00	$3 17	$3 33
2	$1 00	$1 33	$1 67	2 00	2 33	2 67	3 00	3 33	3 67	4 00	4 33	4 67	5 00	5 33	5 67	6 00	6 33	6 67
3	1 50	2 00	2 50	3 00	3 50	4 00	4 50	5 00	5 50	6 00	6 50	7 00	7 50	8 00	8 50	9 00	9 50	10 00
4	2 00	2 67	3 33	4 00	4 67	5 33	6 00	6 67	7 33	8 00	8 67	9 33	10 00	10 67	11 33	12 00	12 67	13 33
5	2 50	3 33	4 17	5 00	5 83	6 67	7 50	8 33	9 17	10 00	10 83	11 67	12 50	13 33	14 17	15 00	15 83	16 67
6	3 00	4 00	5 00	6 00	7 00	8 00	9 00	10 00	11 00	12 00	13 00	14 00	15 00	16 00	17 00	18 00	19 00	20 00
7	3 50	4 67	5 83	7 00	8 17	9 33	10 50	11 67	12 83	14 00	15 17	16 33	17 50	18 67	19 83	21 00	22 17	23 33
8	4 00	5 33	6 67	8 00	9 33	10 67	12 00	13 33	14 67	16 00	17 33	18 67	20 00	21 33	22 67	24 00	25 33	26 67
9	4 50	6 00	7 50	9 00	10 50	12 00	13 50	15 00	16 50	18 00	19 50	21 00	22 50	24 00	25 50	27 00	28 50	30 00
10	5 00	6 67	8 33	10 00	11 67	13 33	15 00	16 67	18 33	20 00	21 67	23 33	25 00	26 67	28 33	30 00	31 67	33 33
11	5 50	7 33	9 16	11 00	12 83	14 67	16 50	18 33	20 17	22 00	23 83	25 67	27 50	29 33	31 17	33 00	34 83	36 67
12	6 00	8 00	10 00	12 00	14 00	16 00	18 00	20 00	22 00	24 00	26 00	28 00	30 00	32 00	34 00	36 00	38 00	40 00
13	6 50	8 67	10 83	13 00	15 17	17 33	19 50	21 67	23 83	26 00	28 17	30 33	32 50	34 67	36 83	39 00	41 17	43 33
14	7 00	9 33	11 67	14 00	16 33	18 67	21 00	23 33	25 67	28 00	30 33	32 67	35 00	37 33	39 67	42 00	44 33	46 67
15	7 50	10 00	12 50	15 00	17 50	20 00	22 50	25 00	27 50	30 00	32 50	35 00	37 50	40 00	42 50	45 00	47 50	50 00
16	8 00	10 67	13 34	16 00	18 67	21 33	24 00	26 67	29 33	32 00	34 67	37 33	40 00	42 67	45 33	48 00	50 57	53 33
17	8 50	11 33	14 17	17 00	19 83	22 67	25 50	28 33	31 17	34 00	36 83	39 67	42 50	45 33	48 17	51 00	53 83	56 67
18	9 00	12 00	15 00	18 00	21 00	24 00	27 00	30 00	33 00	36 00	39 00	42 00	45 00	48 00	51 00	54 00	57 00	60 00
19	9 50	12 67	15 83	19 00	22 17	25 33	28 50	31 67	34 83	38 00	41 17	44 33	47 50	50 67	53 83	57 00	60 17	63 33
20	10 00	13 33	16 67	20 00	23 33	26 67	30 00	33 33	36 67	40 00	43 33	46 67	50 00	53 33	56 67	60 00	63 33	66 67
21	10 50	14 00	17 50	21 00	24 50	28 00	31 50	35 00	38 50	42 00	45 50	49 00	52 50	56 00	59 50	63 00	66 50	70 00
22	11 00	14 67	18 34	22 00	25 67	29 33	33 00	36 67	40 33	44 00	47 67	51 33	55 00	58 67	62 33	66 00	69 67	73 33
23	11 50	15 33	19 17	23 00	26 83	30 67	34 50	38 33	42 17	46 00	49 83	53 67	57 50	61 33	65 17	69 00	72 83	76 67
24	12 00	16 00	20 00	24 00	28 00	32 00	36 00	40 00	44 00	48 00	52 00	56 00	60 00	64 00	68 00	72 00	76 00	80 00
25	12 50	16 67	20 83	25 00	29 17	33 33	37 50	41 67	45 83	50 00	54 17	58 33	62 50	66 67	70 83	75 00	79 17	83 33
26	13 00	17 33	21 67	26 00	30 33	34 67	39 00	43 33	47 67	52 00	56 33	60 67	65 00	69 33	73 67	78 00	82 33	86 67
27	13 50	18 00	22 50	27 00	31 50	36 00	40 50	45 00	49 50	54 00	58 50	63 00	67 50	72 00	76 50	81 00	85 50	90 00
28	14 00	18 67	23 34	28 00	32 67	37 33	42 00	46 67	51 33	56 00	60 67	65 33	70 00	74 67	79 33	84 00	88 67	93 33
29	14 50	19 33	24 17	29 00	33 83	38 67	43 50	48 33	53 17	58 00	62 83	67 67	72 50	77 33	82 17	87 00	91 83	96 67
30	15 00	20 00	25 00	30 00	35 00	40 00	45 00	50 00	55 00	60 00	65 00	70 00	75 00	80 00	85 00	90 00	95 00	100 00

TABLE NO. 42.

HORIZONTAL RETURN TUBULAR BOILERS.

	Horse Power.	Diameter.	Length.	Tubes.		Heating Surface.	Grates.			Plate.		Estimated Weight.			Dome.			Smoke Stack.			
				Number.	Length.		Length.	Width.	Surface.	Shell.	Heads.	Boiler only.	Front and Fixtures.	Total, Boiler and Fixtures.	If Dome is wanted Add to List.	Diam.	Height.	Diam.	Height above Grates in feet.	Thick. of Plate, No. Birmingham Gauge.	Weight with Base.
		in.	ft. in.		ft. in	sq. ft.	ft in.	ft. in.	sq. ft.	in.	in.	lbs.	lbs.	lbs.		in.	in.	in.			
4 Inch Tubes.	35	44	13 6	24	13 6	430	3	3	9	9/32	3/8	5500	4400	9900	$ 44	24	24	18	35	16	750
	40	46	14 6	26	14 6	496	3 6	3	10.5	9/32	3/8	6260	4600	10860	44	24	24	20	40	16	900
	45	48	14 6	30	14 6	560	3 6	3 6	12.25	5/16	7/16	6880	4800	11680	50	26	26	21	50	14	1300
	50	48	16	30	16	618	4	3 6	14	5/16	7/16	7640	4900	12540	50	26	26	21	50	14	1300
	60	54	16 6	34	16 6	724	4	4	16	11/32	1/2	9200	5500	14700	58	28	28	24	60	14	1700
	70	58	15	48	15	890	4 6	4 6	20.25	3/8	1/2	10750	6500	18250	58	28	28	25	60	14	1800
	80	62	16	50	16	933	4 6	5	22.5	3/8	1/2	12230	6700	18930	58	28	28	27	60	12	2500
	90	64	16 6	54	16 6	1098	4 6	5	22.5	3/8	9/16	13190	7400	20590	68	30	30	29	70	12	3000
	100	66	16 6	58	16 6	1174	5	5	25.	13/32	9/16	14810	8000	22810	80	32	32	30	70	12	3300
	110	68	17	66	17	1358	5 6	5 6	30.25	13/32	5/8	16050	8300	24350	94	34	34	30	70	12	3300
	125	72	17	76	17	1548	5 6	6.	33	7/16	5/8	18440	10400	18840	110	36	36	33	75	10–12	3600
	150	72	20	76	20	1816	6	6	36	7/16	5/8	21700	10600	32300	110	36	36	33	80	10–12	3900
	175	74	20	84	20	1992	6 6	6	39	15/32	11/16	24000	11200	35200	130	38	38	36	80	10–12	4300
	200	78	20	88	20	2088	7	6 6	45	1/2	11/16	26900	12600	39500	130	38	38	38	85	10–12	4500
	250	84	20	108	20	2532	8	7	56	9/16	3/4	31100	15000	46100	130	38	38	42	90	10–12	5800

TABLE NO. 43.

HORIZONTAL RETURN TUBULAR BOILERS.

	Horse Power	Diameter.	Length.	Tubes. Number.	Tubes. Length.	Heating Surface.	Grates. Length.	Grates. Width.	Grates. Surface.	Plate Shell.	Plate Heads.	Estimated Weight Boiler only.	Estimated Weight Front and Fixtures.	Estimated Weight Total, Boiler and Fixtures.	Dome, If Dome is wanted Add to List.	Dome, Diam.	Dome, Height.	Smoke Stack. Diam.	Smoke Stack. Height above Grates in feet.	Smoke Stack. Thick. of Plate, No. Birmingham Gauge.	Smoke Stack. Weight with Base.
		in.	ft. in.		ft. in.	sq. ft.	ft in.	ft in.	sq. ft.	in.	in.	lbs.	lbs.	lbs.		in.	in.	in.			
3½ Inch Tubes.	25	42	10 6	28	10 6	338	3	2 6	7.5	¼	⅜	3960	4100	3060	$ 40	22	22	17	35	16	700
	30	44	12	30	12	410	3	3	9	9/32	⅜	4880	4300	9180	44	24	24	18	35	16	750
	35	44	13 6	30	13 6	460	3 6	3	10.5	9/32	⅜	5350	4400	9750	44	24	24	18	35	16	750
	40	46	14 6	30	14 6	498	3 6	3	10.5	9/32	⅜	6000	4600	10600	44	24	24	20	40	16	900
	45	48	14 6	36	14 6	582	3 6	3 6	12.25	5/16	7/16	6800	4800	11600	50	26	26	21	50	14	1300
	50	48	16	36	16	641	4	3 6	14	5/16	7/16	7460	4900	12360	50	26	26	21	50	14	1300
	60	54	16 6	42	16 6	768	4	4	16	11/32	½	9050	5500	14550	58	28	28	24	60	14	1700
	70	58	15 6	54	15 6	904	4 6	4 6	20.25	⅜	½	10440	6500	16940	58	28	28	25	60	14	1800
	80	62	16	62	16	1060	4 6	5	22.5	⅜	½	12100	6700	18800	58	38	22	27	60	12	2500
	90	64	16	70	16	1184	5	5	25.	⅜	9/16	13200	7400	20600	68	30	20	29	70	12	3000
	100	66	16 6	74	16 6	1296	5 6	5	27.5	13/32	9/16	14680	8000	22680	80	32	32	30	70	12	3300
	110	68	16 6	82	16 6	1413	5 6	5 6	30.25	13/32	⅝	15640	8300	23940	94	34	34	30	70	12	3300
	125	72	16 6	94	16 6	1636	6	6	36	7/16	⅝	18280	10400	28680	110	36	36	33	75	10–12	3600
	150	72	19	94	19	1844	6 6	6	39	7/16	⅝	21000	10600	31600	110	36	36	33	80	10–12	3900
	175	74	19 6	100	19 6	2008	6 6	6	39	15/32	11/16	23900	10900	34800	130	38	38	36	80	10–12	4100

TABLE NO. 44.

HORIZONTAL RETURN TUBULAR BOILERS.

	Horse Power.	Diameter.	Length.	Tubes. Number.	Tubes. Length.	Heating Surface.	Grates. Length.	Grates. Width.	Grates. Surface.	Plate. Shell.	Plate. Heads.	Estimated Weight. Boiler only.	Estimated Weight. Front and Fixtures.	Estimated Weight. Total, Boiler and Fixtures.	Dome. If Dome is wanted Add to List.	Dome. Diam.	Dome. Height.	Smoke Stack. Diam.	Smoke Stack. Height above Grates in feet	Smoke Stack. Thick. of Plate, No. Birmingham Gauge.	Smoke Stack. Weight with Base.
		in.	ft. in.		ft. in.	sq. ft.	ft. in.	ft. in.	sq. ft.	in.	in.	l. s	lbs.	lbs.		in.	in.	in.			
3 Inch Tubes.	20	42	8	36	8	280	2 6	2 6	6.25	1/4	3/8	3500	4000	7500	$ 40	22	22	17	35	16	700
	25	42	10	36	10	348	3	2 6	7.5	1/4	3/8	4000	4100	8100	40	22	22	17	35	16	700
	30	44	11	38	11	400	3	3	9	9/32	3/8	4700	4300	9000	44	24	24	18	35	16	750
	35	44	13	38	13	472	3 6	3	10.5	9/32	3/8	5300	4400	9700	44	24	24	18	35	16	750
	40	46	14	44	14	580	3 6	3 6	12.25	9/32	3/8	6200	4600	10800	44	24	24	20	40	16	900
	45	48	14	50	14	652	4	3 6	14	5/16	7/16	6900	4800	11700	50	26	26	21	50	14	1300
	50	48	15	50	15	698	4 6	3 6	15.75	5/16	7/16	7500	4900	12400	50	26	26	21	50	14	1300
	60	54	15 6	62	15 6	882	5	4	20	11/32	1/2	9400	5500	14900	58	28	28	24	60	14	1700
	70	58	15	74	15	1006	5	4 6	22.5	3/8	1/3	10540	6500	17040	58	28	28	25	60	14	1800
	80	62	16	82	16	1182	5	5	25	3/8	1/2	12200	6700	18900	58	28	28	27	60	12	2500
	90	64	16	92	16	1316	6	5	30	3/8	9/16	13300	7400	20700	68	30	30	29	70	12	3000
	100	66	16 6	94	16 6	1386	6	5	30	13/32	9/16	14700	8000	22700	80	32	32	30	70	12	3300
	110	68	16	108	16	1528	6	5 6	33	13/32	5/8	15700	8300	24000	94	34	34	30	70	12	3300
	125	72	16 6	126	16 6	1820	6 6	6	39	7/16	5/8	18400	10400	28800	110	36	36	33	75	10–12	3600
	150	72	18	126	18	1984	6 6	6	39	7/16	5/8	20900	10600	31500	110	36	36	33	80	10–12	3900

TABLE NO. 45.

Pneumatic Tools.

The amount of air necessary to operate various pneumatic tools differs considerably with the various designs and makes, as well as the service to which they are put. It may be generally stated that the hammers take from 15 to 20 cubic feet of free air per minute; drills from 20 to 35 cubic feet, while that for air hoists may be found in the following table:

Diameter Inches	Capacity Pounds	Lift Feet	Cu. Ft. Free Air per 4 Ft. Lift at 80 Lb. Pressure.	Diameter Inches	Capacity Pounds	Lift Feet	Cu. Ft. Free Air Per 4 Ft. Lift at 80 Lb. Pressure
3	470	4	1.17	9	4440	4	10.88
4	930	4	2.13	10	5630	4	13.50
5	1400	4	3.31	12	8015	4	19.58
6	1925	4	4.83	14	10803	4	26.51
7	2660	4	6.63	16	14123	4	34.49
8	3660	4	8.67				

The size and capacity of compressor for furnishing sufficient air have to be decided by the circumstances and conditions of each plant by itself. We find that generally, for the use of pneumatic tools, the capacity of the compressor should be from $\frac{1}{3}$ to $\frac{1}{2}$ the total consumption of all tools, supposing them all to be working at one time.

Cubic Feet of Free Air Per Minute Required to Run a Small Number of Ingersoll-Sergeant Rock Drills, With 60 Pounds Air Pressure at Sea Level.

Number of Machines.	ROCK DRILL — SIZES.								COAL CUTTERS.	
	A 2 in.	B $2\frac{1}{2}$ in.	C $2\frac{3}{4}$ in.	D 3 in.	E $3\frac{1}{4}$ in.	F $3\frac{1}{2}$ in.	G $4\frac{1}{4}$ in.	H 5 in.	$3\frac{1}{2}$ in.	4 in.
1	65	70	95	110	115	125	140	165	70	93
2	110	120	160	190	200	230	250	280	140	186
3	156	174	234	279	294	333	360	405	210	279

NOTE. — In cases where any considerable number of these drills are to be used, the amount of free air required is so much beyond the capacity of our compressors as to require a large number of them, and in such instances, it is more advisable to obtain a larger type of compressor built especially for such purposes.

TABLE NO. 46.

ROCK DRILLS.

Factor for Determining Free Air Per Minute Required for Rock Drills at 60, 70, 80, 90, 100 Pounds Pressure, and Altitudes from Sea Level to 10,000 Feet Above.

ALTITUDE IN FEET ABOVE SEA LEVEL.	ATMOSPHERIC PRESSURE LBS. PER SQR. IN	FACTOR OF MULTIPLICATION. PRESSURE AT DRILL.				
		60 lbs.	70 lbs.	80 lbs.	90 lbs.	100 lbs.
0	14.7	1.00	1.133	1.26	1.40	1.535
500	14.45	1.015	1.15	1.28	1.425	1.563
1,000	14.12	1.03	1.17	1.31	1.43	1.59
1,500	13.92	1.048	1.19	1.33	1.48	1.62
2,000	13.61	1.06	1.21	1.35	1.50	1.645
3,000	13.10	1.10	1.25	1.40	1.55	1.70
4,000	12.61	1.131	1.287	1.443	1.60	1.755
5,000	12.15	1.17	1.33	1.495	1.652	1.81
6,000	11.75	1.20	1.37	1.537	1.705	1.87
7,000	11.27	1.24	1.42	1.59	1.76	1.935
8,000	10.85	1.282	1.465	1.645	1.825	2.00
9,000	10.45	1.32	1.51	1.70	1.90	2.07
10,000	10.10	1.365	1.56	1.755	1.968	2.143

TABLE NO. 47.

Cubic Feet of Air Per Minute Required to Operate a Small Number of Rand Drills of Various Sizes at 60 Pounds Air Pressure at Sea Level.

NO. OR NAME.	KID.	No. 1	No. 2	No. 3	No. $3\frac{1}{4}$	No. 4	No. 5	No. 7
Diam of Cylinder in Inches.	$1\frac{7}{8}$ in.	$2\frac{1}{4}$ in	$2\frac{3}{4}$ in.	$3\frac{1}{8}$ in.	$3\frac{1}{4}$ in.	$3\frac{5}{8}$ in.	$4\frac{1}{2}$ in.	5 in.
Number of Drills.								
1	35	53	64	95	103	112	132	154
2	61	93	112	166	180	196	231	270
3	88	133	160	238	258	280	330	385

TABLE 47.

TABLE OF THE AREAS OF CIRCULAR SEGMENTS FOR DIAMETER = 1.

Height.	Area.	Height.	Area.	Height.	Area.	Height.	Area.
.001	.000 042	.064	.021 168	.127	.057 991	.190	.103 900
.002	.000 119	.065	.021 660	.128	.058 658	.191	.104 686
.003	.000 219	.066	.022 155	.129	.059 328	.192	105 472
.004	.000 337	.067	.022 653	.130	.059 999	.193	.106 261
.005	.000 471	.068	.023 155	.131	.060 673	.194	.107 051
.006	.000 619	.069	.023 660	.132	.061 349	.195	.107 843
.007	.000 779	.070	.024 168	.133	.062 027	.196	.108 636
.008	.000 952	.071	.024 680	.134	.062 707	.197	.109 431
.009	.001 135	.072	.025 196	.135	.063 389	.198	.110 227
.010	.001 329	.073	.025 714	.136	.064 074	.199	.111 025
.011	.001 533	.074	.026 236	.137	.064 761	.200	.111 824
.012	.001 746	.075	.026 761	.138	.065 449	.201	.112 625
.013	.001 969	.076	.027 290	.139	.066 140	.202	.113 427
.014	.002 199	.077	.027 821	.140	.066 833	.203	.114 231
.015	.002 438	.078	.028 356	.141	.067 528	.204	.115 036
.016	.002 685	.079	.028 894	.142	.068 225	.205	.115 842
.017	.002 940	.080	.029 435	.143	068 924	.206	.116 651
.018	.003 202	.081	.029 979	.144	.069 626	.207	.117 460
.019	.003 472	.082	.030 526	.145	.070 329	.208	.118 271
.020	.003 749	.083	.031 077	.146	.071 034	.209	.119 084
.021	.004 032	.084	.031 630	.147	.071 741	.210	.119 898
.022	.004 322	.085	.032 186	.148	.072 450	.211	.120 713
.023	.004 619	.086	.032 746	.149	.073 162	.212	.121 530
.024	.004 922	.087	.033 308	.150	.073 875	.213	.122 348
.025	.005 231	.088	.033 873	.151	.074 590	.214	.123 167
.026	.005 546	.089	.034 441	.152	.075 307	.215	.123 988
.027	.005 867	.090	.035 012	.153	.076 026	.216	.124 811
.028	.006 194	.091	.035 586	.154	.076 747	.217	.125 634
.029	.006 527	.092	.036 162	.155	.077 470	.218	126 459
.030	.006 866	.093	.036 742	.156	.078 194	.219	.127 286
.031	.007 209	.094	.037 324	.157	.078 921	.220	.128 114
.032	.007 559	.095	.037 909	.158	.079 650	.221	.128 943
.033	.007 913	.096	.038 497	.159	.080 380	.222	.129 773
.034	.008 273	.097	.039 087	.160	.081 112	.223	.130 605
.035	.008 638	.098	.039 681	.161	.081 847	.224	.131 438
.036	.009 008	.099	.040 277	.162	.082 582	.225	.132 273
.037	.009 383	.100	.040 875	.163	.083 320	.226	.133 100
.038	.009 764	.101	.041 477	.164	.084 060	.227	.133 946
.039	.010 148	.102	.042 081	.165	.084 801	.228	.134 784
.040	.010 538	.103	.042 687	.166	.085 545	.229	.135 624
.041	.010 932	.104	.043 296	.167	.086 290	.230	.136 465
.042	.011 331	.105	.043 908	.168	.087 037	.231	.137 307
.043	.011 734	.106	.044 523	.169	.087 785	.232	.138 151
.044	.012 142	.107	.045 140	.170	.088 536	.233	.138 996
.045	.012 555	.108	.045 759	.171	.089 288	.234	.139 842
.046	.012 971	.109	.046 381	.172	.090 042	.235	.140 689
.047	.013 393	.110	.047 006	.173	.090 797	.236	.141 538
.048	.013 818	.111	.047 633	.174	.091 555	.237	.142 388
.049	.014 248	.112	.048 262	.175	.092 314	.238	.143 239
.050	.014 681	.113	.048 894	.176	.093 074	.239	.144 091
.051	.015 119	.114	.049 529	.177	.093 837	.240	.144 945
.052	.015 561	.115	.050 165	.178	.094 601	.241	.145 800
.053	.016 008	.116	.050 805	.179	.095 367	.242	.146 656
.054	.016 458	.117	.051 446	.180	.096 135	.243	.147 513
.055	.016 912	.118	.052 090	.181	.096 904	.244	.148 371
.056	.017 369	.119	.052 737	.182	.097 675	.245	.149 231
.057	.017 831	.120	.053 385	.183	.098 447	.246	.150 091
.058	.018 297	.121	.054 037	.184	.099 221	.247	.150 953
.059	.018 766	.122	.054 690	.185	.099 997	.248	.151 816
.060	.019 239	.123	.055 346	.186	.100 774	.249	.152 681
.061	.019 716	.124	.056 004	.187	.101 553	.250	.153 546
.062	.020 197	.125	.056 664	.188	.102 334	....	....
.063	.020 681	.126	.057 327	.189	.103 116	....	....

TABLE 47. — Continued.

TABLE OF THE AREAS OF CIRCULAR SEGMENTS FOR DIAMETER = 1.

Height.	Area.	Height.	Area.	Height	Area.	Height	Area.
.251	.154 413	.314	.211 083	.377	.270 951	.440	.332 843
.252	.155 281	.315	.212 011	.378	.271 921	.441	.333 836
.253	.156 149	.316	.212 941	.379	.272 891	.442	.334 829
.254	.157 019	.317	.213 871	.380	.273 861	.443	.335 823
.255	.157 891	.318	.214 802	.381	.274 832	.444	.336 816
.266	.158 763	.319	.215 734	.382	.275 804	.445	.337 810
.257	.159 636	.320	.216 666	.383	.276 776	.446	.338 809
.258	.160 511	.321	.217 600	.384	.277 748	.447	.339 799
.259	.161 386	.322	.218 534	.385	.278 721	.448	.340 793
.260	.162 263	.323	.219 469	.386	.279 695	.449	.341 788
.261	.163 141	.324	.220 404	.387	.280 669	.450	.342 783
.262	.164 020	.325	.221 341	.388	.281 643	.451	.343 778
.263	.164 900	.326	.222 278	.389	.282 618	.452	.344 773
.264	.165 781	.327	.223 216	.390	.283 593	.453	.345 768
.265	.166 663	.328	.224 154	.391	284 569	.454	.346 764
.266	.167 546	.329	.225 094	.392	.285 545	.455	.347 760
.267	.168 431	.330	.226 034	.393	.286 521	.456	.348 756
.268	.169 316	.331	.226 974	.394	.287 499	.457	.349 752
.269	.170 202	.332	.227 916	.395	.288 476	.458	.350 749
.270	.171 090	.333	.228 858	.396	.289 454	.459	.351 745
271	.171 978	.334	.229 801	.397	.290 432	.460	.352 742
.272	.172 868	.335	.230 745	.398	.291 411	.461	.353 739
.273	.173 758	.336	.231 689	.399	292 390	.462	.354 736
.274	.174 650	.337	.232 634	.400	.293 370	.463	.355 733
.275	.175 542	.338	.233 580	.401	.294 350	.464	.356 730
.276	.176 436	.339	.234 526	.402	.295 350	.465	.357 728
.277	.177 330	.340	.235 473	.403	.296 311	.466	.358 725
.278	.178 226	.341	.236 421	.404	.297 292	.467	.359 723
.279	.179 122	.342	.237 369	.405	.298 274	.468	.360 721
.280	.180 020	.343	.238 319	.406	.299 256	.469	.361 719
.281	.180 918	.344	.239 268	.407	.300 238	.470	.362 717
.282	.181 818	.345	.240 219	.408	.301 221	.471	.363 715
.283	.182 718	.346	.241 170	.409	.302 204	.472	.364 714
.284	.183 619	.347	.242 122	.410	.303 187	.473	.365 712
.285	.184 522	.348	.243 074	.411	.304 171	.474	.366 711
.286	.185 425	.349	.244 027	.412	.305 156	.475	.367 710
.287	.186 329	.350	.244 980	.413	.306 140	.476	.368 708
.288	.187 235	.351	.245 935	.414	.307 125	.477	.369 707
.289	.188 141	.352	.246 890	.415	.308 110	.478	.370 706
.290	.189 048	.353	.247 845	.416	.309 096	.479	.371 705
.291	.189 956	.354	.248 801	.417	.310 082	.480	.372 704
.292	.190 865	.355	.249 758	.418	.311 068	.481	.373 704
.293	.191 774	.356	.250 715	.419	.312 055	.482	.374 703
.294	.192 685	.357	.251 673	.420	.313 042	.483	.375 702
.295	.193 597	.358	.252 632	.421	.314 029	.484	.376 702
.296	.194 509	.359	.253 591	.422	.315 017	.485	.377 701
.297	.195 423	.360	.254 551	.423	.316 005	.486	.378 701
.298	.196 337	.361	.255 511	.424	.316 993	.487	.379 701
.299	.197 252	.362	.256 472	.425	.317 981	.488	.380 700
.300	.198 168	.363	.257 433	.426	.318 970	.489	.381 700
301	.199 085	.364	.258 395	.427	.319 959	.490	.382 700
.302	.200 003	.365	.259 358	.428	.320 949	.491	.383 700
.303	.200 922	.366	.260 321	.429	.321 938	.492	.384 699
.304	.201 841	.367	.261 285	.430	.322 928	.493	.385 699
.305	.202 762	.368	.262 249	.431	.323 919	.494	.386 699
.306	.203 683	.369	.263 214	.432	.324 909	.495	.387 699
.307	.204 605	.370	.264 179	.433	.325 900	.496	.388 699
.308	.205 528	.371	.265 145	.434	.326 891	.497	.389 699
.309	.206 452	.372	.266 111	.435	.327 883	.498	.390 699
.310	.207 376	.373	.267 078	.436	.328 874	.499	.391 699
.311	.208 302	.374	.268 046	.437	.329 866	.500	.392 699
.312	.209 228	.375	.269 014	.438	.330 858	....	....
.313	.210 155	.376	.269 982	.439	.331 851	....	....

HARTFORD SPECIFICATIONS.

The following are the specifications, for steel plate, of the Hartford Steam Boiler Inspection and Insurance Co.

OPEN HEARTH FIRE BOX STEEL

To have a tensile strength of not less than 55,000 lbs., nor more than 62,000 lbs. per square inch of section, with not less than 56% of ductility as indicated by contraction of area at point of fracture under test and by an elongation of 25% in a length of 8 inches.

HEADS—To be made of best Open Hearth Flange Steel, 60,000 T. S. All plates, both of shell and heads, must be plainly stamped with name of maker, brand and tensile strength; brands so located that they may be seen on each plate after boiler is finished. Each shell plate must bear a coupon which shall be sheared off, finished up and tested by the maker of the boiler, at his own expense. Each coupon must fill the above requirements as to strength and ductility, and must also stand bending down double when cold, when red hot and after being heated red hot and quenched in cold water, without signs of fracture. All plates failing to pass these tests will be rejected. All tests and inspections of material shall be made at the place of manufacture prior to shipment.

TABLE NO. 48.

Showing details of rivet laps for different thicknesses of boiler plate as advocated by the Hartford Steam Boiler Inspection and Insurance Co. for

DOUBLE RIVETED BUTT JOINTS.

Thickness of Plate.	Diameter of Rivets.	Pitch of Rivets in inches.	Width of Outside Butt Strap.	Width of Inside Butt Strap.	Thickness of Covering Straps.	Vertical or Transverse Pitch.	Edge of Butt Strap to Center of Rivets.	Pitch of Rivets Girth Seam.	Edge of Plate to Center of Rivets Girth Seam.	Strength of Joint.
5/16 in	11/16 in	2 1/4 x 4 1/2	4 1/2 in	9 in	1/4 in	2 1/4 in	1 1/8 in	2 1/8 in	1 1/8 in	83%
3/8 "	3/4 "	2 3/4 x 4 3/4	4 7/8 "	9 7/8 "	5/16 "	2 7/16 "	1 1/4 "	2 1/8 "	1 7/32 "	82.9%
7/16 "	13/16 "	2 15/32 x 4 15/16	5 1/4 "	10 1/2 "	3/8 "	2 3/8 "	1 15/16 "	2 1/4 "	1 7/32 "	82%
1/2 "	7/8 "	2 9/16 x 5 1/8	5 7/8 "	11 1/4 "	7/16 "	2 13/16 "	1 13/32 "	2 1/4 "	1 13/32 "	80%

TABLE NO. 48.—Continued.

TRIPLE RIVETED BUTT JOINTS.

Thickness of Plate.	Diameter of Rivets.	Pitch of Rivets in inches.	Width of Outside Butt Strap.	Width of Inside Butt Strap.	Thickness of Covering Straps	Vertical or Transverse Pitch.	Edge of Butt Strap to Center of Rivets.	Pitch of Rivets Girth Seam.	Edge of Plate to Center of Rivets Girth Seam.	Strength of Joint.
1/4 in	1/4 in	2 1/4 x 4 1/2	6 1/2 in	11 3/8 in	1/4 in	1 1/2 in	27/32 in	2 1/16 in	1 1/8	87.5%
9/32 "	9/16 "	2 9/32 x 4 9/16	6 3/4 "	12 3/8 "	1/4 "	1 1/2 "	15/16 "	2 1/16 "	1 1/8	86%
5/16 "	11/16 "	3 1/8 x 6 1/4	9 1/4 "	14 "	1/4 "	2 1/8 "	1 1/4 "	2 1/16 "	1 1/8	88%
11/32 "	11/16 "	3 1/8 x 6 1/4	9 1/4 "	14 "	1/4 "	2 1/8 "	1 1/4 "	2 1/8 "	1 7/32	88%
3/8 "	3/4 "	3 1/4 x 6 1/2	9 3/4 "	14 1/4 "	6/16 "	2 3/16 "	1 7/32 "	2 1/8 "	1 7/32	87.5%
13/32 "	3/4 "	3 1/4 x 6 1/2	9 1/4 "	14 1/4 "	6/16 "	2 3/16 "	1 7/32 "	2 1/8 "	1 7/32	87.5%
7/16 "	7/8 "	3 3/8 x 6 3/4	10 1/8 "	15 5/8 "	3/8 "	2 1/4 "	1 13/32 "	2 7/16 "	1 1/8	86%
15/32 "	7/8 "	3 1/2 x 7	10 3/8 "	16 "	13/32 "	2 3/8 "	1 13/32 "	2 7/16 "	1 1/8	86%
1/2 "	15/16 "	3 3/4 x 7 1/2	11 "	16 3/4 "	7/16 "	2 3/8 "	1 17/32 "	2 1/8 "	1 1/8	86.6%

For detailed drawings of above laps, see pages 946 to 951 inclusive.

STAYING BOILER HEADS.

In the return tubular boiler the tubes occupy a considerable portion of the boiler head, usually amounting to about two-thirds of the entire area of the head, as shown by Figure 403. There

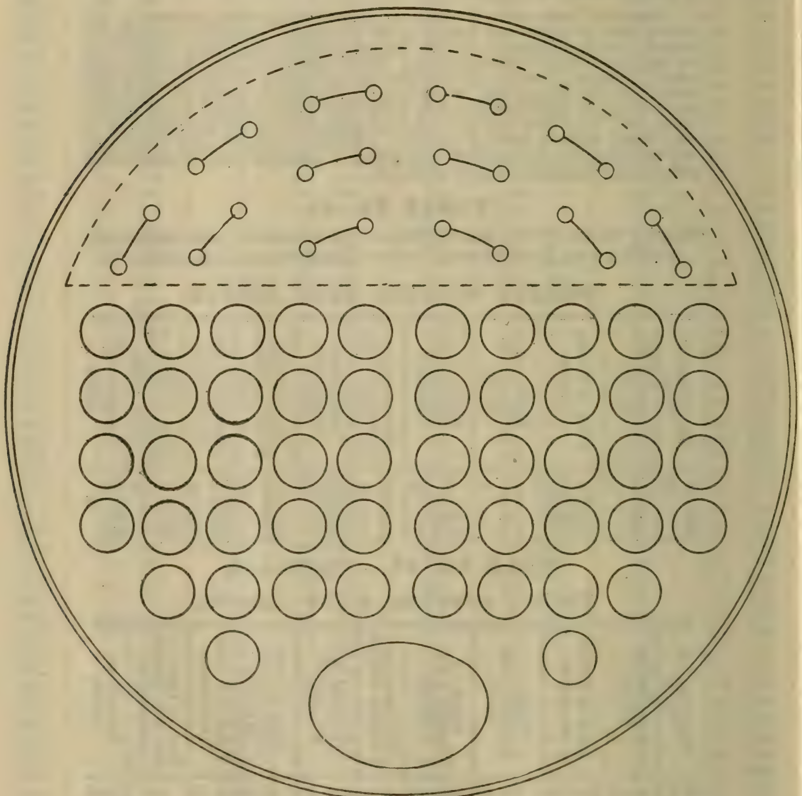


Fig. 403.

Showing No. of stays in a well proportioned boiler of 60 in. in diameter.

is very little surface, comparatively speaking, between the tubes for pressure to act upon, and as the tubes serve as stays to a con-

siderable extent, it is, therefore, unnecessary to introduce stays between them. The remaining flat surface above and below the tubes is exposed to full boiler pressure, and would tend to bulge outward, and thus loosen the tube ends if not properly secured in position. The upper part of the head can be held in position by means of rods connected to both heads, or by means of stays connected to the head and boiler shell. When the area above the tubes is secured by means of rods connected to both heads, the rods are called direct, or through stays, and when the rods run from the head to the shell, they are called diagonal stays.

There is no necessity for using direct or head-to-head stays in the return tubular boiler, because the shell possesses a surplus of strength in the direction of its length, which is sufficient to resist all the pressure that can be brought to bear on the area of the head above and below the tubes. The ability of the shell to aid in strengthening the head can be made use of by putting in the proper number of diagonal stays. The advantages offered by the latter style of stay are that the stays occupy comparatively little space in the steam room of the boiler, thus permitting the interior, both above and below the tubes, to be thoroughly inspected and cleaned, and the stays being shorter, the effects of expansion are less noticeable upon the boiler heads, and the tendency of the stays to work loose is correspondingly reduced.

When calculating the number of stays required in a boiler head, it is not necessary to include the entire area above the tubes, because it has been found by experiment that the flange of the head imparts sufficient strength to a distance of 3 inches from the flange, and that the tubes tend to stay the head to a distance of 2 inches above the tops of the upper row of tubes. It will be seen, therefore, that the actual area to be stayed extends to within 3 inches of the shell, and to within 2 inches of the top of the tube. The area to be stayed is contained within

the dotted lines shown in Fig. 404, and is in the form of a segment of a circle.

In order to find the total pressure or stress on this area it is first necessary to find the area of the segment formed by the dotted lines. It will be seen that the top of this segment is an

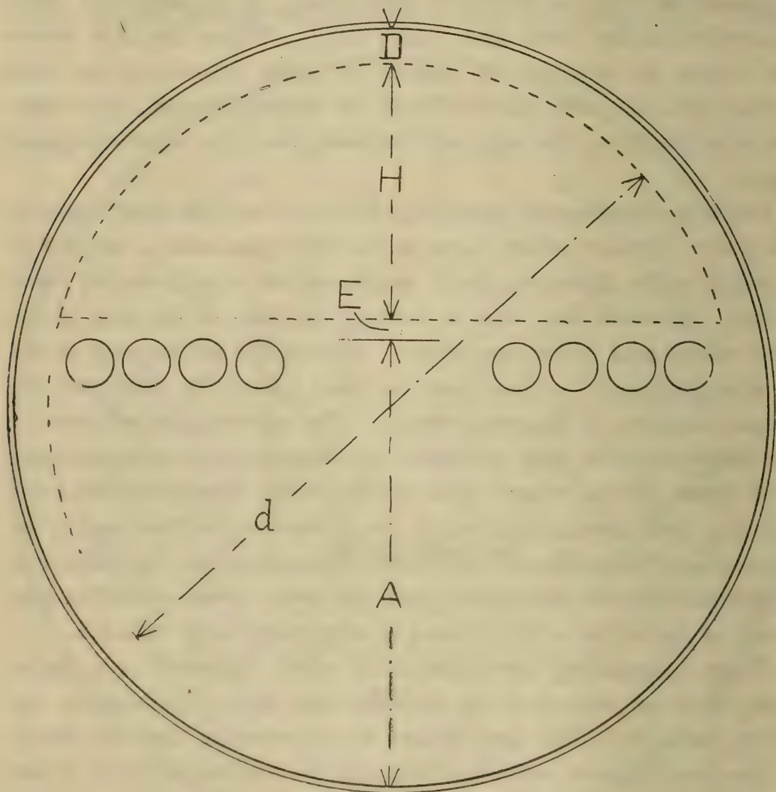


Fig. 404.

Showing the area to be stayed.

arc of a circle which is smaller than the boiler head. If we take off 3 inches from the diameter of the head at both top and bottom it will give the diameter of the circle of which the segment is

a part, and the diameter will be found to be 6 inches less than the diameter of the boiler head. The distance, H, Fig. 404, is called the height of the segment. The area of the segment multiplied by the steam pressure gives the total stress to be resisted by the stays, and the total stress divided by the stress which one stay will hold gives the number of stays required.

To illustrate the method of finding the number of stays required, suppose the stays in a 60-inch and a 72-inch boiler head are to be determined. In a well proportioned boiler head the distance from the bottom of the shell to the top of the upper row of tubes is about 60 per cent of the diameter of the head, and 60 per cent of 60 inches is 36 inches, which is the distance, A, Fig. 404. The distance from the top of the tubes to the top of the boiler head is $60 - 36 = 24$ inches. According to the rule previously given we must subtract the width of the space, D, which is three inches, and also the width of the space, E, which is 2 inches thus making the height of the segment equal to $24 - (3 + 2) = 19$ inches, which is the height, H, Fig. 404. Divide the height, H, by the diameter, d, of the circle of which the segment is a part. In the table of areas of segments of circles on page 935, find the quotient in the column headed, Height, and multiply the corresponding decimal in the column headed area by the square of the dameter of the circle of which the segment is a part; the product will be the area of the segment included within the dotted lines. Thus, the diameter of the circle of which the segment is a part is $60 - 6 = 54$ inches, and $19 \div 54 = .351$. In the column headed, Height, we find .351 and opposite we find the decimal .2459, which, when multiplied by the square of the diameter of the circle, gives $.2459 \times (54 \times 54) = 717$ square inches, the area of the segment. If the boiler pressure is to be 100 pounds on each square inch the total pressure will be $717 \times 100 = 71{,}700$ pounds, which is the stress to be borne by all the stays. When stays are not subjected to the action of water they may be

allowed a stress of 7,500 pounds per square inch of section, but when subjected to the action of the water in the boiler the stress should not exceed 6,000 pounds.

Now stays in horizontal tubular boilers are generally made of 1 inch round bars and as a bar of this size has an area of .7854 of a square inch each bar or stay will stand a stress of $7,500 \times .7854 = 5890$ pounds, and the number of stays required is $71,700 \div 5890 = 12$ stays, as shown in Fig. 403.

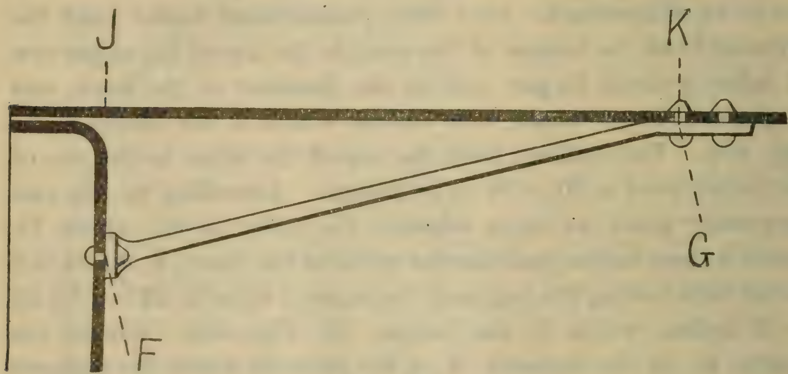


Fig. 405.

Method of connecting a diagonal stay.

The stress on diagonal stays is a little greater than on direct stays running from head to head, so that the area of the 12 stays must be increased in proportion to the increased stress. The minimum length of stay allowable is $3\frac{1}{2}$ feet, and the stress on the diagonal stay is as much greater than the stress on the direct stay as the length of the stay, FG, Fig. 405, is greater than the distance JK. Suppose the distance, JK, is 45 inches, and the length of the stay, FG, 48 inches, then the stress on the diagonal stay is $48 \div 45 = 1\frac{7}{100}$ times the stress on the direct stay and consequently the area of the stay must be $1\frac{7}{100}$ times the area calculated or $.7854 \times 1\frac{7}{100} = .84$ of a square inch, which corresponds to a diameter of $1\frac{1}{16}$ inches. Thus the 60-inch boiler requires 12 stays each $1\frac{1}{16}$ inches in diameter, as shown in Fig. 403.

The number of stays required in the 72-inch boiler is found in the same manner. Thus, the diameter of the circle of which the segment to be braced is a part, is 72 — 6 = 66 inches. The distance from the bottom of the shell to the top of the tubes is 60 per cent of 72 inches, or 43 inches, and the distance from the top of the tubes to the top of the shell is 72 — 43 = 29 inches. From this is to be subtracted 3 inches measured from the shell, and 2 inches measured from the tubes, making 5 inches in all, thus leaving 29 — 5 = 24 inches, which is the height, H, of the segment represented by the dotted lines. Dividing the height of the segment by the diameter of the circle of which the segment is a part, we have 24 ÷ 66 = .363. In the table of areas of segments of circles find .363 in the column headed, Height, and opposite we find the decimal .2574, which, when multiplied by the square of the diameter of the circle of which the segment is a part, gives .2574 × (66 × 66) = 1121 square inches area of segment. Assume the pressure to be 110 pounds per square inch. The total stress to be borne by all the stays is 1121 × 110 = 123,310 pounds. If the stays are to be of 1 inch rods, which is the smallest size used for boilers above 44 inches in diameter, the area of each rod or stay will be .7854 of a square inch, and it will be capable of resisting a stress of 7,500 × .7854 = 5890 pounds.

The number of 1 inch stays required, therefore, is 123,310 ÷ 5890 = 21 stays.

Now, if the distance, J K, Fig. 405, in this case is 48 inches, and the length of the stay 52 inches, the stress on the diagonal stay will be 52 ÷ 48 = $1\frac{8}{100}$ times what it would be on a direct stay, and consequently the area of the stay must be $1\frac{8}{100}$ times greater, or .7854 × 1.08 = .85 square inch, which corresponds to a diameter of $1\frac{1}{16}$ inches, therefore the 72-inch boiler will require 21 stays $1\frac{1}{16}$ inches in diameter.

It sometimes happens that the maximum pressure employed

when calculating stays calls for a certain number which it is found difficult to properly distribute over the boiler head. In this case the number of stays must be changed, but the combined area of the stays must necessarily remain the same. To illustrate, sup-

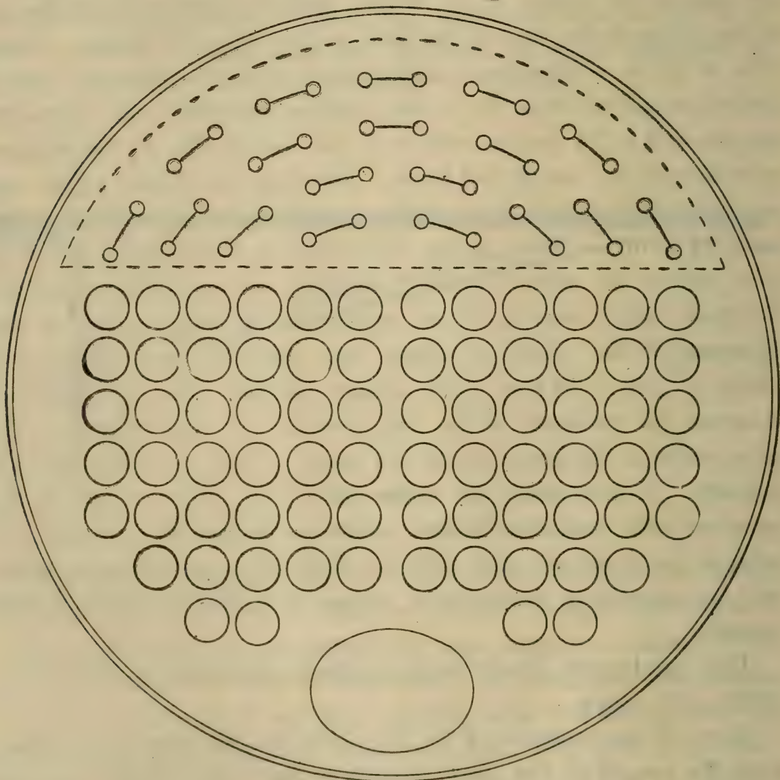


Fig. 406.

No. of stays in a well proportioned boiler of 72 in. in diameter.

pose 21 stays could not be distributed to advantage on the 72-inch head, and it is desired to use, say, 18 stays. Now, the 21 stays have a combined area of $.85 \times 21 = 17\frac{7}{8}$ square inches, and this area divided among 18 stays would give each an area of

$17\frac{7}{8} \div 18 = 1$ square inch, which corresponds to a diameter of $1\frac{1}{8}$ inches, therefore the 72-inch boiler will require 18 stays $1\frac{1}{8}$ inches in diameter.

The same method is used when finding the number and size of the stays below the tubes. It is seldom practicable, with the usual arrangement of tubes and man-hole, to use more than two stays below the tubes so that the area is made sufficient to enable two stays to carry the stress.

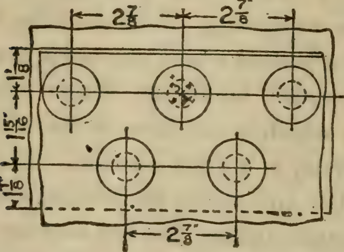
When finding the size and number of screw stays or stay bolts, such are used in the water-legs of locomotive boilers and in the headers of water tube boilers where the bolts are subjected to the action of the water, the maximum stress allowable per square inch of net section, which is the area of the bolt measured at the bottom of the threads, is 6,000 pounds. For instance, what is the diameter, number and stress on the staybolts required in the water leg of a locomotive boiler, the flat surface measuring 40×54 inches, and the maximum pressure 125 pounds? The total area is found to be $40 \times 54 = 2,160$ square inches, and the total pressure to be resisted by the staybolts is $2,160 \times 125 = 270,000$ pounds. For pressures exceeding 100 pounds per square inch, the pitch of staybolts in stationary boilers should not exceed $4\frac{1}{4}$ inches. The area supported by each bolt is $4\frac{1}{4} \times 4\frac{1}{4} = 18$ square inches and the stress on each bolt is $18 \times 125 = 2,250$ pounds. The total stress is 270,000 pounds, consequently 270,000 divided by 2,250 will give the number of bolts which is $270,000 \div 2,250 = 120$. The stress allowed per square inch of net sectional area is 6,000 pounds, and as each bolt must sustain a stress of 2,250 pounds the net area of each bolt is $2,250 \div 6,000 = .375$ or $\frac{3}{8}$ of a square inch measured at the bottom of the threads, which corresponds to a bolt $\frac{3}{4}$ inch in diameter measured to outside of threads. This surface, therefore, requires 120 staybolts $\frac{13}{16}$ inch in diameter, spaced $4\frac{1}{4}$ inches between centers for a working pressure of 125 pounds.

DOUBLE RIVETED LAP AND GIRTH JOINTS

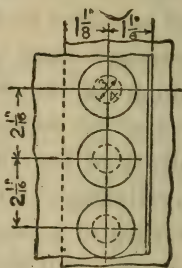
Longitudinally Riveted.)

This construction is based on a tensile strength of 60,000 lbs. for plate and a shearing strength of 38,000 lbs. for rivets.

LAP JOINT.

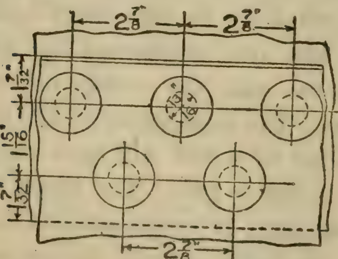


GIRTH JOINT.

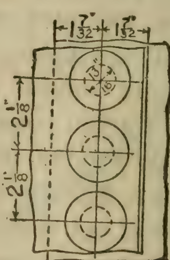


¼" plates, double riveted; holes, ¾"; rivets, 11/16". (Efficiency, 74%.)

LAP JOINT.



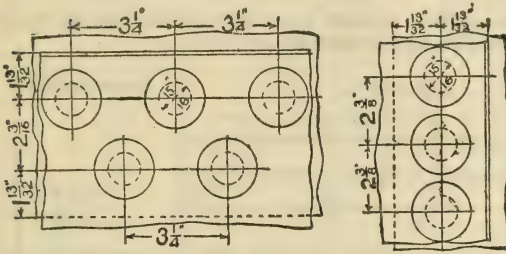
GIRTH JOINT.



5/16" plates, double riveted; holes 13/16"; rivets, ¾". (Efficiency, 72%.)

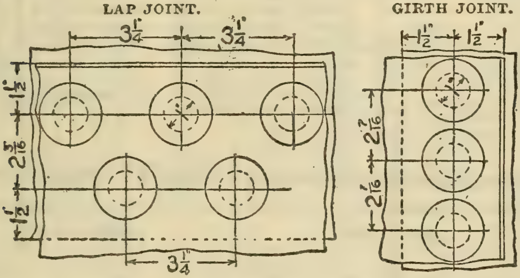
Double Riveted Lap and Girth Joints.

LAP JOINT. GIRTH JOINT.



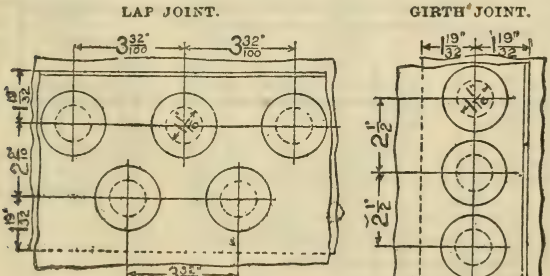
⅜" plates, double riveted; holes, 15/16"; rivets, ⅞". (Efficiency, 70%.)

LAP JOINT. GIRTH JOINT.



7/16" plates, double riveted; holes, 1"; rivets, 15/16". (Efficiency, 70%.)

LAP JOINT. GIRTH JOINT.



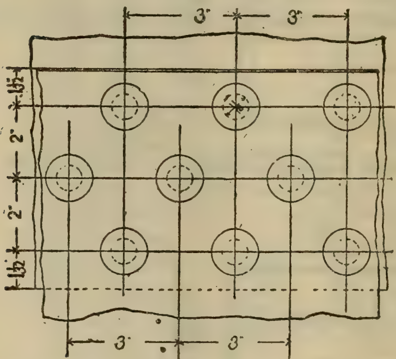
½" plates, double riveted; holes, 1 1/16"; rivets, 1" (Efficiency, 68%.)

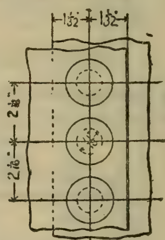
Triple Riveted Lap and Girth Joints.

(Longitudinally Riveted.)

This construction is based on a tensile strength of 60,000 lbs. for plate and a shearing strength of 38,000 lbs. for rivets.

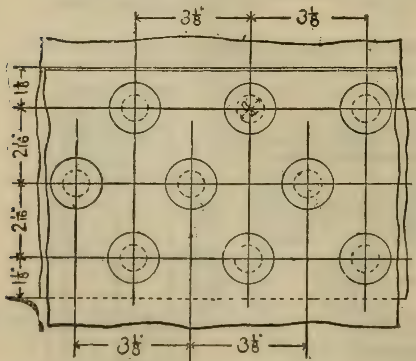
LAP JOINT. GIRTH JOINT.

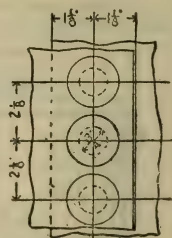




$\frac{1}{4}$" plates, triple riveted; holes, $\frac{11}{16}$": rivets, $\frac{5}{8}$". (Efficiency, 77%.)

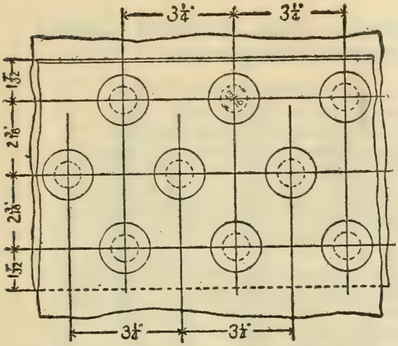
LAP JOINT. GIRTH JOINT.

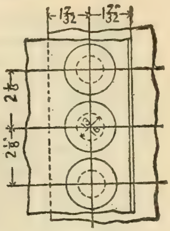




$\frac{5}{16}$" plates, triple riveted; holes, $\frac{3}{4}$"; rivets, $\frac{11}{16}$". (Efficiency, 76%.)

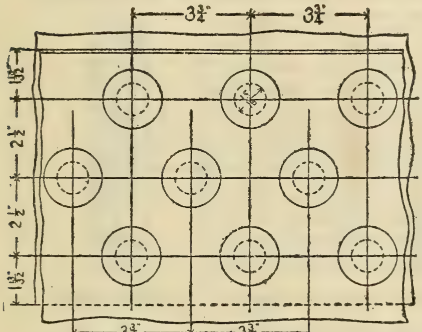
LAP JOINT. GIRTH JOINT.

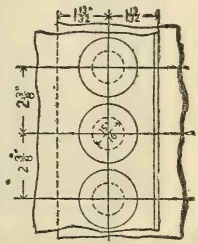




$\frac{3}{8}$" plates, triple riveted; holes, $\frac{13}{16}$"; rivets $\frac{3}{4}$". (Efficiency, 75%.)

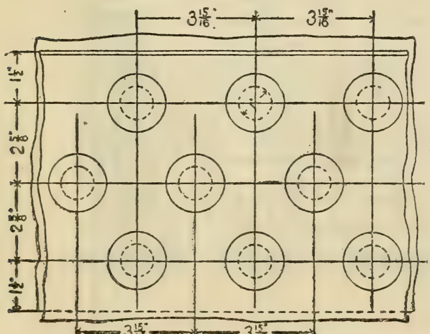
LAP JOINT. GIRTH JOINT.

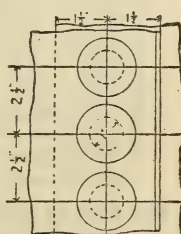




$\frac{7}{16}$" plates, triple riveted; holes, $\frac{15}{16}$"; rivets, $\frac{7}{8}$". (Efficiency, 75%.)

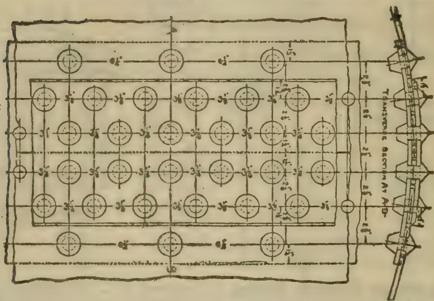
LAP JOINT. GIRTH JOINT.



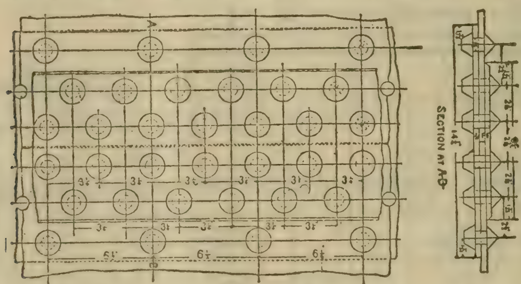


$\frac{1}{2}$" plates, triple riveted; holes, 1"; rivets, $\frac{15}{16}$". (Efficiency, 75%.)

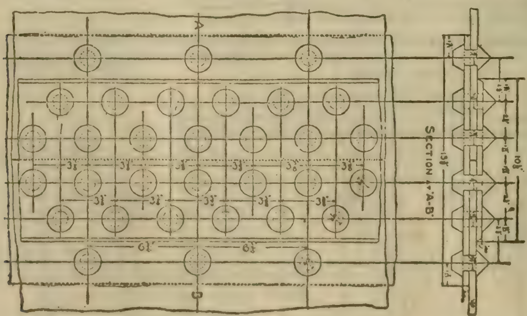
Triple Riveted Butt Joints



Butt joint for $\frac{5}{16}$" plates; holes, ¾"; rivets, $\frac{11}{16}$". (Efficiency, 88%.)

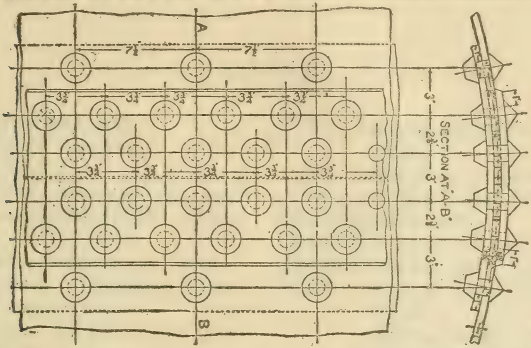


Butt joint for ⅜" plates; holes, $\frac{13}{16}$"; rivets, ¾". (Efficiency, 87½%.)

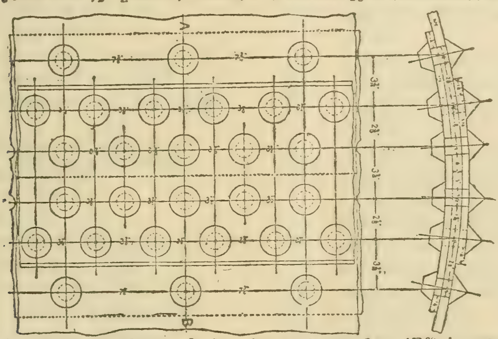


Butt joint for $\frac{7}{16}$" plates; holes, $\frac{15}{16}$;" rivets, ⅞". (Efficiency, 86%.)

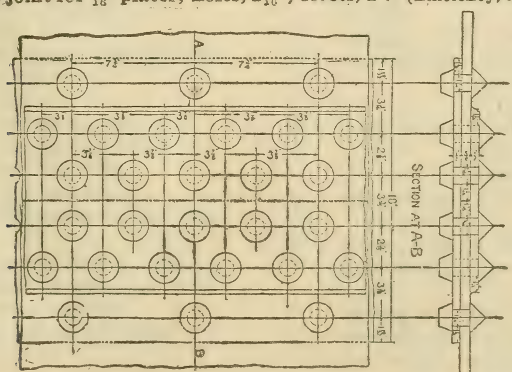
TRIPLE RIVETED BUTT JOINTS—(Continued.)



Butt joint for ½" plates; holes, 1"; rivets, $\frac{15}{16}$" (Efficiency, 86.6%.)



Butt joint for $\frac{9}{16}$" plates; holes, $1\frac{1}{16}$"; rivets, 1". (Efficiency, 86%.)



Butt joint for ⅝" plates; holes, $1\frac{1}{16}$" rivets 1". (Efficiency, 86%.)

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